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CONTROL OF NOISE AND SONIC VIBRATION IN SHIPS

by Igor' Ivanovich Klyukin

- USSR -

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CONTROL OF NOISE AND SONIC VIBRATION IN SHIPS

- USSR -

This publication contains the complete translation of the Russian-language book by Igor' Ivanovich Klyukin, Bor'ba s shumom i zvukovoy vibratsiyey na sudakh (Control of Noise and Sonic Vibration in Ships); Editorial Board: Prof. Dr. of Physico-Mathematical Sciences S. N. Rzhevkin, Engineer I. Ya. Kolesnikov and M. S. Antsyferov. Publishing House of Shipbuilding Literature (Sudpromgiz), Leningrad, 1961, 355 pages. The book was signed to the press 8 September 1961, 2800 copies.

(Note: The attention of the reader is invited to the statement of the author in Section 4 of this book that one Russian bar is equal to one international microbar.)

TABLE OF CONTENTS

Table of Contents (continued) Page

PART II

THE ISOLATION AND ABSORPTION OF NOISE AND SONIC VIBRATION IN SHIPS

Table of Contents (continued) Page

PREFACE

Discussed are,

~~This book discusses~~ the sources of noise in ships and ~~explains~~ the bases for controlling noise and vibration in ships. A short treatment of the fields of physical, physiological and structural acoustics, as well as acoustical measurements, is presented. Modern methods of sound and vibration insulation, and sound and vibration absorption, and methods for measuring these values are described. The causes of the origin of vibrations in machinery and mechanisms are analysed, and several means and methods for controlling these vibrations are indicated. The material discussed is illustrated by examples and calculations.

The book is intended for use by designers, scientific personnel, shipbuilding personnel and the manufacturers of shipboard machinery. It may be utilised also by students of marine and architectural acoustics, acoustics of mechanisms, and of several technical applications of the theory of vibrations. Certain chapters of the present book may be utilised by persons concerned with controlling noise and vibration in other transport vehicles, in buildings and in industry.

"Sometime in the future mankind will succeed in controlling noise as decisively as it has succeeded in controlling cholera and bubonic plague."

Robert Koch (1843-1910)

(Epigram of the exhibit "Less Noise!" at the Third International Congress on Acoustics, September, 1939.)

INTRODUCTION

1938 -- 68 phons
1952 -- 77 phons
1957 -- 80 phons

These figures, like the epigram above, were taken from the data of the Third International Congress on Acoustics. They indicate the average annual increase in noisiness of transport vehicles. An increase of 12 phons in the noise level (i.e., a more than two-fold increase in the level of noise) over a period of some 18 years, as indicated by world statistics, is a value sufficient to evoke a pause for reflection.

Ships do not constitute an exception in this respect. The extensive introduction of Diesel and gas turbine equipment. the increase in their power and pressure, and the increased speed of [illegible] all result in an increase in the noise level of the service, passenger and navigation areas of ships. Thus it is quite natural that control of noise and vibration, aimed at improving the habitability of ships, has acquired prime importance during recent years.

The control of noise and lessening of the effect of noise and vibration of man is not a new subject. The classical works of Helmholz, Rayleigh, G. Lamb, A. N. Krylov and S. P. Timoshenko form the scientific basis of this field of technical acoustics. Considerable research in the field of sound absorption, and in the related fields of physical, physiological, architectural and mensural acoustics have been performed by the native acoustics scientists N. N. Andreyev, A. I. Belov, L. M. Brekhovskikh, V. S. Grigor'yev, S. N. Rzhevkin, I. I. Slavin, V. V. Furduyev, Ye. Ya. Yudin and others.

Of the research conducted abroad mention may be made of the fundamental investigations of G. Fletcher and G. Bekeshi on the physiology of hearing, the works of F. Morse, E. Meyer, L. Kremer, L. Beranek, V. Knudsen, K. Gesel, U. Ingard, F. Ingerslev, G. Oberst, V. Tseller and others, on the absorption and insulation of sound.

The majority of the work done on sound shielding has dealt with the problems connected with lessening noise in industrial, housing and public buildings, and in surface and air transport vehicles. The field of shipboard acoustics has found relatively weak expression in the scientific and technical literature.

Shipboard acoustics are distinguished from architectural acoustics by several peculiarities. Firstly, these have abundant metallic structures, through which sound may penetrate to remote areas; there is considerable saturation of areas with powerful and noisy machines, such as diesels, ventillators and pumps, which as a rule have variable schedules of operation; heavy sound-shielding structures used in stationary construction cannot be utilized. The peculiarities listed above determine a certain specific approach to acoustic problems of ships. Attention has been focused most constantly in this respect on the problems of propagation of vibration, vibration insulation and absorption, and direct action upon the sources of sound, and the problems of development of light, and sufficiently efficient sound shielding means.

In connection with domestic investigations of acoustics having direct bearing on ships and other means of transportation, it would be pertinent to mention the works of Ye. Ts. Andreyeva-Galinina on the effects of vibration on man, M. S. Antsyferov on the absorption of the sound of shipboard ventillators and on vibrometry, N. N. Babayev on shipboard vibration, N. G. Belyakovskiy on vibration-insulating structures, M. D. Genkin, V. I. Zinchenko and K. I. Selivanov on reduction of the noise of shipboard mechanisms, Yu. I. Iorish on vibrometry, B. D. Tartakovskiy on vibration-absorbing coatings, and V. P. Terskikh on the vibrations of ships' propeller shafting. These and other works, investigations and projects conducted in the USSR by institutes of the Academy of Sciences and of the industries and plant laboratories create a definite basis for the formulation of shipboard acoustics as one of the branches of structural and architectural acoustics on one hand, and of the structural mechanics of shipbuilding on the other.

The present book is the first attempt at a generalization of the results of investigations and projects in the field of controlling sound and vibrations in the sonic frequency band in ships. In addition to the data of Russian and foreign investigators, the book contains a summary of the author's experiences in relation to various problems of shipboard acoustics.

The book consists of three parts. The first, or introductory part presents the necessary information from the fields of physical, physiological and mensural acoustics; the sources and level of noise

on shipboard are indicated, and a general classification of the methods of controlling sound and sonic vibrations is given.

The second, and main part of the book is devoted to exposition of the methods of improving the acoustic characteristics of ships' compartments. The sequence of chapters of this part conforms to the foregoing classification, i. e., the chapters dealing with control of airborne noise by the methods of sound isolation and sound absorption are first, followed by the chapters dealing with control of sonic vibrations by vibration isolation and absorption.

Problems in quieting shipboard equipment and mechanisms at the source are discussed in the third part of the book, although it would appear more natural to discuss them on the basis of the means of sound-shielding of compartments. This is explained by the fact of the present insufficiency of experience in the field of elimination of the noise of machinery at the source, and by the fact that the means of isolation and absorption utilized in elimination of the noise of these mechanisms are described in part two. Yu. S. Kryuchkov participated in drawing up part three, and the author expresses his gratitude to him.

The material present in the book is illustrated by numerical and graphic examples. All computations in the examples were performed with a level of precision equal to that of a 25-centimeter slide rule. In most cases the mathematical values are expressed in the CGS system, the use of which enables employment of the Russian standard of acoustic values (GOST 8849-58, dated 1 January 1959).

The author is indebted to L. Ya. Gutin, who reviewed the manuscript, to the critics S. N. Rshevkin and I. Ya. Kolesnikov, and to Editor M. S. Antsyferov for their very valuable notations. Comments of readers relative to the present book also will be received with gratitude.

The Author

PART I

BRIEF SUMMARY OF PHYSICAL AND PHYSIOLOGICAL ACOUSTICS AND ACOUSTICAL MEASUREMENTS

CHAPTER 1

GENERAL CONCEPTS AND DEFINITIONS OF PHYSICAL ACOUSTICS

1. Vibration Systems and Vibratory Motion. Oscillation of Systems With Concentrated Constants

Any mechanical system having elements of elasticity and mass may produce vibrations upon application of a periodic force to the system. If the periodic exciting factor is present during the entire time of vibration it is called a forced vibration. If the system is excited from a state of equilibrium and vibrates under the effect of its own internal forces the motion is referred to as free vibration. Free vibrations are brought about by the transition of kinetic energy to potential energy, and back to kinetic. Kinetic energy is accumulated by the elements of mass, and potential energy is accumulated by the elastic elements.

The simplest example of a vibrating system is a concentrated weight suspended by a spring, one end of which is fixed to a support (Figure 1). In a system of this kind the elements of elasticity and mass are separated from each other, and it is called a system with concentrated constants. In acoustics we encounter also systems with distributed constants, in each portion of which elastic elements are combined with mass elements.

In the system illustrated in Figure 1, no external force is applied in its free oscillation, and thus the internal forces acting in the system, elastic and inertial forces, are equilibrated. According to Newton's second law the inertial force is equal to the product

of the mass M and its acceleration, and the elastic force is proportionate to the displacement of the spring.

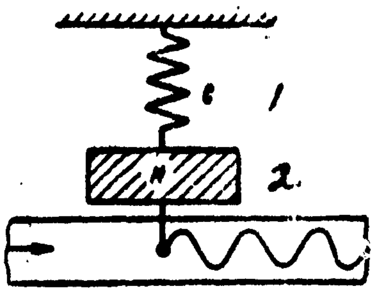


Figure 1. Simplest vibrating system and its vibration curve.

1. Spring 2. Mass

The equation of forces acting within the system at any given moment has the form:

$$M\ddot{y} + Cy = 0; \qquad (1)$$

where C is the elasticity or rigidity of the spring, computed as the force which must be applied to the spring to effect unit displacement;

y momentary value of the oscillatory displacement of the spring;

$\ddot{y}$ momentary value of the oscillatory acceleration of the mass, equal to the second derivative of the oscillatory displacement according to time.

The simplest form of resolution of the differential equation of the free oscillations of a system with concentrated constants, having one degree of freedom (represented by the coordinate y in the given case), is a harmonic function of time. The fact that a harmonic, i.e., sinusoidal or cosinusoidal function satisfies the equation of oscillatory movement may be verified by fastening a stylus to the moving mass. A sinusoid wave is drawn on paper moving perpendicularly to the direction of oscillation (Figure 1).

Equation (1) also satisfies an exponential function of an imaginary argument which may be based on the mathematical formula of Euler and

expressed as the sum or difference of two harmonic functions. In the following we shall frequently undertake resolution of equations of oscillations in the form of exponential functions, expressing the solution of equation (1) in the simplest form:

$$y = y_a \sin \omega t; \qquad (2)$$

where y_a - is the amplitude of the oscillatory displacement. In Figure 2, depicting the formation of a curve of harmonic oscillation, y_a represents the amplitude of the rotating radial vector, the projection of which on the vertical axis gives a sinusoid function;

t - is time;

ω - is the rotational velocity, indicating the angle (in radians) traversed by the radial vector per unit time.

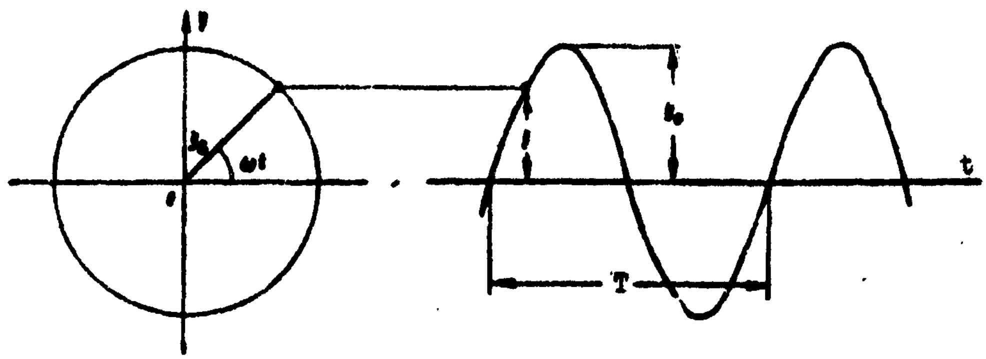


Figure 2. Determining the elements of a harmonic oscillatory process.

A very important characteristic of oscillatory motion is the period of oscillation, i.e., the time of one complete oscillation (indicated by T). It is apparent, that:

$$\omega T = 2\pi,$$

from which we have

$$T = \frac{2\pi}{\omega}. \qquad (3)$$

Very often in technical acoustics not the period of oscillation, but its reciprocal value, the frequency f of oscillation, is used, indicating the number of complete oscillations per second. The frequency is expressed in cycles, and thus is equal to:

$$f = \frac{1}{T} = \frac{\omega}{2\pi} . \tag{4}$$

An expression of the momentary value of the speed of oscillation of a mass may be obtained from equation (2). This speed is equal to the first derivative of oscillatory displacement with time:

$$\dot{y} = \frac{\partial y}{\partial t} = y_a \omega \cos \omega t = \dot{y}_a \sin\left(\omega t + \frac{\pi}{2}\right), \tag{5}$$

where

$$\dot{y}_a = y_a \omega. \tag{6}$$

As we have seen, the amplitude of the speed of oscillation $\dot{y}_a$ is ω-fold greater than the amplitude of oscillatory displacement. The argument of the trigonometric function in the expression $\dot{y}$ is distinguished from the corresponding argument y by $\frac{\pi}{2}$. This implies a phase displacement of the vector of the speed of oscillation relative to the vector of oscillatory displacement equal to a 90° angle.

The speed of oscillation of the mass is equal to:

$$\ddot{y} = \frac{\partial^2 y}{\partial t^2} = -y_a \omega^2 \sin \omega t = \ddot{y}_a \sin \omega t, \tag{7}$$

where

$$\ddot{y}_a = -y_a \omega^2. \tag{8}$$

The amplitude of the vibrational acceleration numerically exceeds the amplitude of oscillatory displacement ω^2-fold. The minus sign indicates that the direction of the vector of the acceleration is opposite to the direction of the vector of oscillatory displacement, i.e., that in the very simple oscillatory system considered the phase displacement between the acceleration and displacement of oscillations is equal to 180°.

Example 1. Given are two vibrating plates. The speed of oscillation of the first plate is $f_1 = 20$ cycles per second, and its amplitude of oscillatory displacement is $y_{a1} = 0.5$ mm. The second plate oscillates with a speed of $f_2 = 1{,}000$ cycles per second, and its amplitude is $y_{a2} = 0.0005$ mm. The problem is to determine the acceleration of oscillation of both plates.

Solution. On the basis of formulas (8) and (4) the amplitude of the acceleration of oscillation of the first plate is

$$\ddot{y}_{a1} = (2\pi f)^2 y_{a1} = (2\pi \cdot 20)^2 \cdot 0.05 = 790 \text{ cm/sec}^2.$$

The acceleration of oscillation of the second plate is

$$\ddot{y}_{a2} = (2\pi \cdot 1000)^2 \cdot 0.00005 = 1970 \text{ cm/sec}^2.$$

Thus the acceleration of oscillation of the second plate is 2.5-fold greater than that of the first, although the amplitude of its oscillation is 1,000-fold less than that of the first.

From this example it is not difficult to deduce a practical conclusion relative to instruments for measuring vibration, viz. instruments reacting to oscillatory acceleration must be much more sensitive in the field of high frequency than the instruments for measuring the speed of oscillation, which in turn must be more sensitive than instruments reacting to the magnitude of oscillatory displacement.

We turn next to expression for the frequency of free oscillation of the system depicted in Figure 1, i.e., the frequency with which this system oscillates when it is removed from an equilibrium situation and oscillates by itself. Inserting expression (7) and (2) into formula (1), we obtain, after reduction of the value of rotational frequency of the free oscillations of the system (indicating this frequency by ω_0):

$$\omega_0 = \sqrt{\frac{C}{M}}. \qquad (9)$$

Taking account of the relationship between ω and f (formula (4)), we obtain an expression of the frequency of free oscillations f_0 of the system:

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{C}{M}} = \frac{1}{2\pi}\sqrt{\frac{C}{G}g}. \qquad (10)$$

where G is the weight of the oscillating mass;

g is the acceleration due to gravity.

Example 2. A machine of weight $G = 200$ kg is set on four shock-absorber springs; each of the springs is able to compress through a distance of 1 mm under the effect of a force of 10 kg. The problem is to determine the frequency of free oscillations of the mechanism on the shock absorbers. By what factor should the number of shock absorbers be increased to effect a two-fold increase in this frequency?

Solution. The magnitude of f_0 may be found according to formula (10), expressing all magnitudes in a system such as that of CGS. In practice, however, it is more convenient to express the rigidity in kg/cm, and the weight in kg. In this, g in formula (10) will be equal to 981 cm/sec^2.

The elasticity of each of the shock absorbers is:

$$C_1 = \frac{10}{0.1} = 100 \text{ kg/cm}.$$

The frequency of free oscillations of the shock absorbing mechanism is:

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{4\cdot 100}{200}981} \approx 7 \text{ cycles}.$$

To increase the frequency two-fold the number of shock absorbers must be increased 4-fold.

2. Forced Oscillation of a System With Concentrated Constants, Mechanical Resistance, Resonance of Oscillations

The foregoing section presented a brief survey of free vibrations of very simple oscillatory systems with concentrated constants. When some external periodic activating force acts on the system the oscillation of the system becomes forced. The equation of forced oscillation is obtained from equation (1), supplementing the right-hand side of the equation with a term characterising the exciting force.

To the left side of the equation we also add a term characterising the reaction of the force of friction, which is present to greater or lesser degree in oscillatory systems. In engineering calculations the force of friction most often is taken as proportionate

to the speed of oscillation (viscous friction). Adoption of the law of viscous friction greatly simplifies analysis, although this law by no means is always true in actual oscillating systems.

Thus the equation of the vibrations of a very simple oscillatory system with viscous friction may be written as follows:

$$M\ddot{y} + R\dot{y} + Cy = F, \tag{11}$$

where R is the coefficient of friction;

F is the exciting force.

The periodical exciting force may be expressed in the form of a harmonic function or exponential function, i.e., in the form:

$$F = F_a e^{j\omega t} \tag{12}$$

where F_a is the amplitude of the force;

e is the natural logarithm base $e = 2.718$;

$j = \sqrt{-1}$

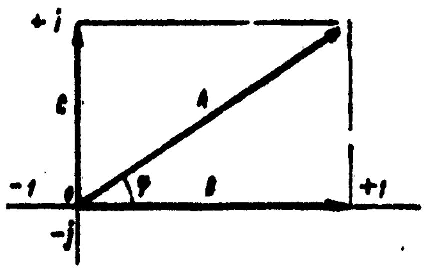


Figure 3. Representation of a vector and its components in a complex plane.

The representation of oscillating forces and parameters of oscillatory processes (displacement, speed, acceleration) in the form of exponential functions derives from the symbolic method, which is applied extensively in acoustics and electrotechnology. At the basis of this method is the utilization of geometric interpretation of complex numbers, i.e., representation of vectors as the sum of two of their projections, on real and imaginary coordinate axes (Figure 3). For example, vector A in Figure 3 may be expressed in the following form:

$$A = B + jC \tag{13}$$

where B and C are the real and imaginary components of the vector, i.e., its projections on the corresponding coordinate axes of a complex plane.

From Figure 3 it may be seen that

$$\left.\begin{array}{l} B = |A| \cos\varphi ; \\ C = |A| \sin\varphi ; \end{array}\right\} \qquad (14)$$

where $|A|$ is the absolute value, or module of the complex number comprising vector A;

φ is the phase angle (complex number argument).

It is readily seen that the value of $|A|$ is equal to

$$|A| = \sqrt{B^2 + C^2}. \qquad (15)$$

Therefore the vector A may be represented in the form:

$$A = |A| (\cos\varphi + j \sin\varphi). \qquad (16)$$

This is the so-called trigonometric form of depicting a complex number. Taking into account the relationship between trigonometric and exponential functions known from the field of mathematics, the value A may be represented also in exponential form:

$$A = |A| e^{j\varphi} \qquad (17)$$

The latter form of depiction of a complex number is utilized in acoustics. The reader who must acquaint himself in greater detail with complex numbers and their geometric representation or desires to refresh his memory with respect to this field of mathematics may refer to the corresponding mathematics textbook or handbook (26, 97). The problems connected with the application of the symbolic method in acoustics and the theory of oscillations are discussed in the books of F. Morse (76), V. V. Furduyev (105) and G. Olson (82). From these works it may be seen that this method is very suitable for resolution of vibration problems which at first glance appear to be affected by several different conditions.

The value j occasionally is called the phase-rotational operator. For the purpose of clarification of the reason and essence of this appellation we must turn to the expression of oscillatory displacement, oscillation speed and the acceleration of oscillation of mass in the system with one degree of freedom as described in Section 1 of the present chapter (formulas (2), (5) and (8)). Expressing the oscillatory displacement in exponential form, we have:

$$y = y_a e^{j\omega t} \tag{18}$$

The speed of oscillation of the mass is:

$$\dot{y} = \frac{\partial y}{\partial t} = j\omega y_a e^{j\omega t} = j\omega y. \tag{19}$$

If the vector of oscillatory displacement is projected upon the real axis, on the side of the positive value of y, the vector of the speed of oscillation will be projected upon the imaginary axis, i.e., axis j (Figure 4). Thus multiplication by j in the formula (19) would indicate a rotation of 90° in the vector.

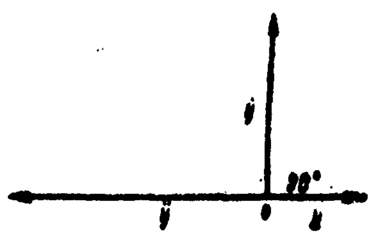


Figure 4. Vector diagram of oscillatory displacement, speed and acceleration in a frictionless system.

The oscillatory acceleration of the mass is equal to:

$$\ddot{y} = \frac{\partial^2 y}{\partial t^2} = j\omega y_a e^{j\omega t} \cdot j\omega = -\omega^2 y. \tag{20}$$

The vector of oscillatory acceleration is projected on the material axis on the side of the negative value of y, i.e., forms an angle of 180° with the displacement vector. This may be considered formally as a consequence of the rotational multiplication by j. On the whole, the vector diagram of displacement, speed and acceleration (Figure 4) is, as may be desired, similar to a diagram which could be constructed according to formulas (5) and (8), obtained with the aid of trigonometric functions.

Let us turn again to equation (11). The general solution of this equation contains two terms: the first term corresponds to the free oscillations of the system, which in the given case are diminishing, due to the presence of friction in the system; the second corresponds to forced oscillation. The expression of oscillatory displacement under forced oscillation may take the form of formula (18). Inserting the expressions of displacement, speed and acceleration from formulas (18) - (20), and expression F from formula (12) into formula (11), we obtain, after reduction of the exponential factor:

$$-M\omega^2 y_a + j\omega R y_a + C y_a = F_a, \tag{21}$$

from which we have

$$y_a = \frac{F_a}{C - M\omega^2 + j\omega R}. \tag{22}$$

For determination of the amplitude of the oscillatory displacement it is necessary to make use of the modulus of the complex expression (22). On the basis of formula (15) the modulus y_a is equal to:

$$|y_a| = \frac{F_a}{\sqrt{(C - M\omega^2)^2 + (\omega R)^2}}. \tag{23}$$

Let us now find the value of the vector and the modulus of the amplitude of the speed of oscillation. Knowledge of this value is very important because the value of the speed of oscillation, and not oscillatory displacement, determines the amount of energy radiated during vibration, and the degree of sound isolation of the structure.

In determining the value of the speed of oscillation we express the value of oscillatory displacement and acceleration through the value of the speed of oscillation. From formulas (19) and (20) we have:

$$y = \frac{\dot{y}}{j\omega} = -j\frac{\dot{y}_a e^{j\omega t}}{\omega}; \quad \ddot{y} = j\omega\dot{y} = j\omega\dot{y}_a e^{j\omega t}. \tag{24}$$

Following insertion in formula (11) and reduction, we have:

$$j\omega M\dot{y}_a + R\dot{y}_a - j\frac{C}{\omega}\dot{y}_a = F_a. \tag{25}$$

Whence

$$\dot{y}_a = \frac{F_a}{j\left(\omega M - \frac{C}{\omega}\right) + R}; \tag{26}$$

$$|\dot{y}_a| = \frac{F_a}{\sqrt{\left(\omega M - \frac{C}{\omega}\right)^2 + R^2}}. \tag{27}$$

The expression in the denominator of formula (26) is called the total mechanical resistance of the system (occasionally it is called the impedence of the system). Indicating this by the symbol z, we have:

$$z = j\left(\omega M - \frac{C}{\omega}\right) + R = \frac{F_a}{\dot{y}_a}, \tag{28}$$

i.e., the total mechanical resistance of the system is equal to the ratio of vibratory force acting in the system to the speed of oscillation evoked by this force. The dimension of the mechanical resistance is dynes·sec/cm.

Let the oscillatory system consist of a single mass only, i.e. $C = 0$ and $R = 0$. Then from formula (28) we have:

$$z = z_M = j\omega M = \frac{F_a}{\dot{y}_a}. \tag{29}$$

From this it is apparent that the mechanical resistance of the mass to the established oscillatory motion is proportional to the frequency of oscillation. The mechanical resistance of the elements of elasticity and friction have similar form

$$z_c = -j\frac{C}{\omega} = \frac{C}{j\omega}; \tag{30}$$

$$z_T = R. \tag{31}$$

The mechanical resistance of the elastic element is inversely proportionate to the frequency of oscillation, but the element of viscous friction does not depend upon frequency. Furthermore, from formulas (29) through (31) it is apparent that the resistance of mass and elasticity are plotted on the imaginary axis (with the former in the positive direction, and the latter in the negative direction), but the resistance of friction is plotted on the real axis. Multiplication by j, as described in the above, corresponds to a phase rotation of 90°. Thus the mass effects a phase shift of the speed of oscillation of 90° relative to the oscillatory force, and the elasticity effects a shift of 90° in the opposite direction. With the action of a single frictional force, only, there is no phase shift between the oscillatory force and the speed of oscillation.

The frictional resistance, evoking an irreversible loss of energy, is called an active resistance. The resistance of mass and elasticity, evoking only the appearance of a phase shift between force and speed, but no energy loss, is called a reactive resistance.

Figure 5 depicts the frequency dependence of the amplitudes of the mass and elasticity components of mechanical resistance, i.e., the values

$$|z_M| = \omega M, \qquad -|z_C| = -\frac{C}{\omega}.$$

It may be noted that in distinction from Figures 3 and 4, the curves of Figure 5 are plotted not on a complex plane, but in an ordinary coordinate system, in which the values of the modulus of the vector of mechanical resistance are plotted on the vertical axis (with observance of sign) and the values of the frequency of oscillation are plotted on the horizontal axis.

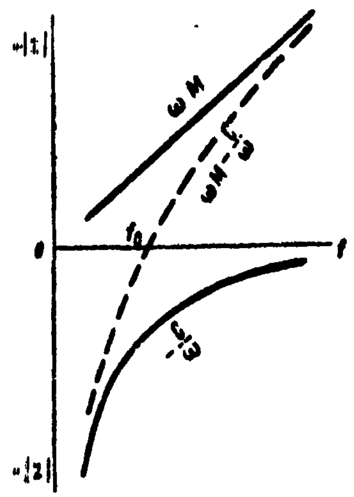


Figure 5. Frequency function of the components of mechanical resistance in an oscillatory system, without resistance.

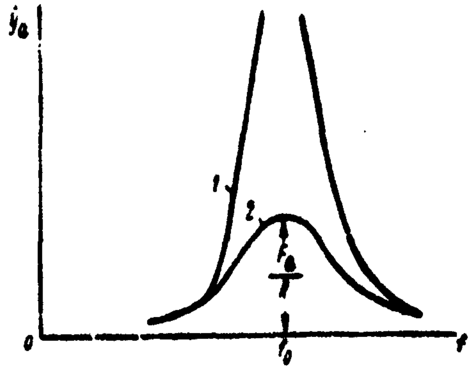


Figure 6. Resonance curves. 1 - In a system without friction; 2 - In a system with friction

At a definite frequency the total resistance of a system without resistance (broken-line curve) progresses through zero. This is the frequency of resonance of the system. For a circular resonance frequency ω_0, therefore,

$$\omega_0 M = \frac{C}{\omega_0}.$$

Thence the resonance frequency of a system without friction is:

$$f_0 = \frac{\omega_0}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{C}{M}}. \qquad (32)$$

The resonance frequency of a system of this kind is found to be equal to the frequency of free oscillation of the system (formula 10). The frequency of resonant oscillation in a system with friction is somewhat lower than the frequency of natural oscillations in a system without friction; however, in the overwhelming majority of cases of practical interest this difference may be neglected.

At the resonant frequency of a system its reactive resistance is equal to zero. The speed of oscillation of the mass is equal to (from formula 26):

$$\dot{y}_{ares} = |\dot{y}_{ares}| = \frac{F_a}{R}. \qquad (33)$$

In the absence of the force of friction from the system the speed of oscillation at resonant frequency increases without limit (Figure 6, curve 1). The greater the mechanical loss in the system, the weaker is the appearance of resonance (curve 2).

<u>Example 3</u>. In the system depicted in Figure 1 the weight is equal to 3 kg, and the stiffness of the spring is 300 kg/cm. The problem is to determine the magnitude and character of the mechanical resistance of the system at frequencies 20 cycles above and below the resonance frequency.

<u>Solution</u>. The resonant frequency of the system is:

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{C}{G}g} = \frac{1}{2\pi}\sqrt{\frac{300\cdot 981}{3}} \approx 50 \text{ cycles.}$$

The magnitude of the mechanical resistance of the system at a frequency of 70 cycles is:

$$z_{70} = j\left(2\pi f M - \frac{C}{2\pi f}\right) = j\left(2\pi\cdot 70\cdot 3\cdot 10^3 - \frac{300\cdot 10^6}{2\pi\cdot 70}\right) \approx$$

$$\approx j(13.2 - 6.8)\cdot 10^5 = j\cdot 6.4\cdot 10^5 \text{ dynes sec/cm.}$$

The resistance of the system at frequencies above the resonant frequency are inertial in character (as was seen from equation (29), the inertial character of a system is reflected by the sumbol $+j$ before the value of the resistance).

At a frequency of 30 cycles the mechanical resistance of the system is equal to:

$$z_{30} = j\left(2\pi\cdot 30\cdot 3\cdot 10^{3} - \frac{300\cdot 10^{6}}{2\pi\cdot 30}\right) \simeq j(5.65-15.9)\cdot 10^{5} =$$
$$= -j\cdot 10.2\cdot 10^{5} \text{ dynes sec/cm.}$$

At a frequency below resonance the resistance has an elastic character (the sign $-j$ before the resistance value, cf. equation (30)).

Example 4. In the system characterising the data of the previous example, the element of friction has been added. The frictional resistance is $R = 10^5$ dynes·sec/cm. The problem is to determine the ratio of the amplitude of the speed of oscillation of the system at resonance, to the amplitude of the speed of oscillation at a frequency of 30 cps. [cycles per second].

Solution. Because the frictional force is relatively small, the resonant frequency of the system is not significantly different from the frequency of free oscillations of a system without friction, but the resistance of the system at a frequency of 30 cycles is practically equal to its reactive resistance, determined in the preceding example.

From formulas (33) and (27) it follows that at a constant exciting force the ratio of the amplitude of the speed of oscillation at resonant frequency to the amplitude of the speed of oscillation at any other frequency is equal to the ratio of the moduli of the mechanical resistances of the system at these frequencies. The mechanical resistance of the system at resonant frequency is determined by the magnitude of the friction value (formula 31). The magnitude of mechanical resistance of the system at a frequency of 30 cps. was determined in the previous example. Using these values, we have:

$$\frac{|\dot{y}_{a\,res}|}{|\dot{y}_{a\,30}|} = \frac{10.2\cdot 10^{5}}{10^{5}} = 10.2.$$

The amplitude of the speed of oscillation at resonance is 10-fold greater than the amplitude of the speed of oscillation at a frequency differing by only 20 cycles from the resonant frequency.

The sharp increase in the speed of oscillation at resonance is explained by the relatively small value of friction in the system.

The phenomenon of resonance plays an important role in structural acoustics. Resonance may be observed not only in vibrations of a machine as a whole on elastic supports, but also in the elastic material of anti-vibration mounts, just as plates, bars, air buffers and air spaces. Oscillating systems either radiate or transmit the greatest amount of oscillatory energy at resonance. However, the phenomenon of resonance may be utilised for the opposite purpose, for absorption of vibration.

3. Origin of Sound, Speed of Propagation of Sound

Elastic and inertial forces exist not only in systems with concentrated constants, but also in solid media. These forces derive from a correspondingly elastic interaction of particles of the medium and the inertial properties of the mass of the particles. The elasticity and mass seem to be distributed throughout the elements of the medium, and because of this a solid medium is called a system with distributed constants. Elastic vibration also is possible in them, consisting of a series of subsequent compressions and rarefactions, propagating from the source of excitation with a definite velocity. This process of propagation of mechanical vibrations in a medium is called sound.

In particular, the sound process arises in the medium surrounding the system of Figure 1, through the oscillation of the latter. The back-and-forth motion of the mass leads to the creation of compression and rarefaction of the medium at its surface, which propagates through the surrounding mass of medium and is perceived as sound (Figure 7).

Not only various mechanical oscillatory systems with concentrated constants may serve as sound generators, but also vortical foci, cavitation bubbles (in a liquid), and surfaces which rub against each other. Similar processes of the generation of sound vibrations in mechanisms are discussed in Part III.

Sound propagating through air is called airborne sound. Sound propagating through sufficiently extensive solid bodies may be called material, or structural sound. In solid bodies of finite dimensions (plates, bars) the sound process appears in the form of sonic vibrations.

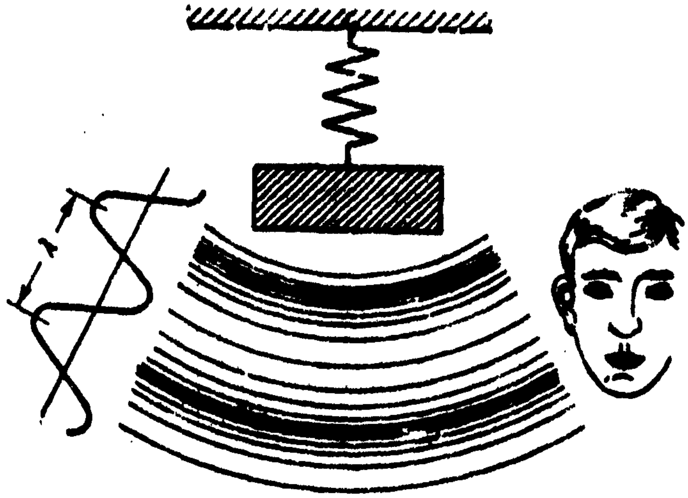


Figure 7. Formation of sound from the vibrations of a system with concentrated constants.

Reviewing the picture of propagation of sound vibration in any given medium it may be noted that in the direction of propagation there are points with identical degree of compression or rarefaction and that these are located at identical distances (Figure 7). The distance between these points, having identical phase of oscillation, is called the length of the wave. It may be noted also, that the length of the wave is the distance travelled by a sound wave during a certain period. If the wave length is indicated by the symbol λ and the speed of propagation of the vibration in the medium is indicated by c, we have the relationship

$$\lambda = \frac{c}{f} = cT \tag{34}$$

where f and T, as in the case of an oscillating system with concentrated constants, are the frequency and period of vibration.

The speed of propagation of sound or, as it is abbreviated, the speed of sound, is a very important characteristic of the sound process. The magnitude of this speed depends upon the character of the medium and the form of propagation of sound waves in the medium.

Gaseous and liquid media are entirely characterized by one elastic constant, the coefficient of compressibility (or the reciprocal of its value, the modulus of elasticity). In these media only one type of sound waves may occur, compression, or longitudinal waves, i.e.,

those in which the direction of vibration of particles of the medium coincides with the direction of propagation of the waves. Let us imagine a conventional system with distributed constants in the form of a series of parallel linkages of an infinite number of alternating elements of elasticity and mass. In the case of longitudinal oscillations we have displacement of the masses and deformation of the springs, transmitted from one element to another, in the direction of the length of the linkage (one of these linkages is depicted in Figure 8).

Homogeneous isotropic solid media are characterized by two elastic constants. Most often we find the modulus of elasticity (Young's modulus) and the modulus of shear as these constants. The presence of a second elastic constant corresponds to the possibility of the appearance of deformation of shear, as well as deformation of compression, in unbounded solid media, with the result that two types of waves exist, longitudinal and transverse. Transverse waves are distinguished from longitudinal waves in that the vibrations are perpendicular to the propagation of the wave (Figure 8). Other types of waves also are possible in bounded solid bodies and at the boundaries of media.

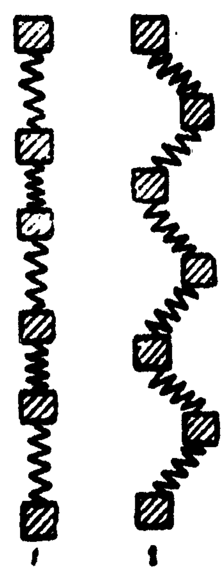


Figure 8. Longitudinal (1), and transverse (2) vibration of particles of a continuous medium.

Let us consider the expressions of the speed of sound in various media. In a gaseous medium the speed of sound is equal to:

$$c = c_{gas} = \sqrt{\frac{p_0}{\rho}\psi}; \qquad (35)$$

where ρ and p_0 are the density and static pressure of the gas;

ψ is the ratio of the specific heat of the gas at constant pressure to the specific heat at constant volume (in the case of air, $\psi = 1.4$).

The speed of sound in a gas depends upon the temperature. In air this dependent function has the form:

$$c_{air} = 332 + 0.6t \text{ m/sec} \qquad (36)$$

where t is the temperature, in degrees Centigrade.

At temperatures in the range of 20°C the speed of sound in air is close to 340 m/sec. The dependence of the speed of sound upon temperature is weaker in the exhaust gases of engines, containing a considerable amount of carbon oxides, than in air.

The speed of sound in liquids is determined according to the formula:

$$c_{liquid} = \sqrt{\frac{K}{\rho}}, \tag{37}$$

where K is the so-called modulus of volumetric elasticity.

In practical computations the speed of sound in water and liquid fuels may be taken as approximately 1,500 m/sec.

As mentioned in the foregoing, two types of waves exist in unbounded solid media. The formula for calculating the speed of propagation of longitudinal waves is:

$$c_1 = \sqrt{\frac{E}{\rho}}\sqrt{\frac{1-\mu}{(1-2\mu)(1+\mu)}}, \tag{38}$$

where E is Young's modulus;

μ is Poisson's ratio;

ρ is the density of the material of the medium.

In metals c_1 has a value of 3 to 5 km/sec. In media with a large coefficient of loss due to internal friction (rubber, plastics, wood), the value of the speed of propagation of vibrations is a function of the frequency of the vibrations.

The values of the speed of propagation of longitudinal waves in several materials and media of major importance are given in the several columns of Table 1. The table also presents the density values of these materials and media necessary for practical computations.

In thin bars, the transverse dimensions of which are less than the length of longitudinal waves, the speed of propagation of longitudinal waves is equal to:

$$c_{bar} = \sqrt{\frac{E}{\rho}}. \tag{39}$$

If the transverse dimensions of the bar are several-fold greater than the length of the sound waves, the speed of propagation of longitudinal vibrations in the bar may be considered equal to the speed of propagation of these vibrations in an unbounded medium (formula 38).

As may be seen from formulas (38) and (39), the speed of propagation of longitudinal waves in thin bars is distinguished from the speed of propagation of the same waves in unbounded media or very wide bars by the absence of the factor containing Poisson's ratio. This difference is not very significant in metals due to the relatively small value of Poisson's ratio. In rubber and several rubber-like materials used for isolation of sound vibration Poisson's ratio is approximately 0.5, and because the denominator of formula (38) contains the value $1 - 2\mu$, the speed of sound in unbounded media of these materials may be several-fold greater than the speed of sound in bars.

Table 1.

DENSITY AND SPEED OF PROPAGATION OF LONGITUDINAL SONIC WAVES, AND THE ACOUSTIC RESISTANCE OF SEVERAL MEDIA AND MATERIALS

Medium, Material	Density, g/cm^3	Speed of Propagation of Sound, m/sec	Specific Acoustic Resistance, $g/sec \cdot cm^2$
Air	0,0012	340	41
Water	1	1500	$1,5 \cdot 10^5$
Steel	7,85	5000	$3,9 \cdot 10^6$
Aluminum	2,7	5100	$1,3 \cdot 10^6$
Rubber (bars)	1—2	40—160	$4 \cdot 10^3 - 3,2 \cdot 10^4$
Cork	0,2	430—530	$8,6 \cdot 10^3 - 1,2 \cdot 10^4$
Wood (pine)			
with the grain	0,7	3400	$2,4 \cdot 10^5$
across the grain	0,7	2500	$1,7 \cdot 10^5$
Wood (fir)			
with the grain	0,5	5200	$2,5 \cdot 10^5$
across the grain	0,5	2400	$1,2 \cdot 10^5$
Glass	2,5	5200	$1,3 \cdot 10^6$

Example 5. The transverse dimensions of the rubber part of a sound-isolating rubber mount are several times greater than the length of longitudinal waves in the rubber. The problem is to find the ratio of the velocity of longitudinal waves in such a broad mount to that

in a mount with a small load-carrying area of rubber. Poisson's ratio for the rubber is $\mu = 0.49$.

Solution. The speed of sound in the mount with a small isolating surface in relation to the wave length may be considered as equal to the speed of sound in thin bars, because the lateral surface of the coating may expand laterally. In a broad rubber coating the propagation of these waves approximates their speed of propagation in an unbounded medium. Utilizing formulas (38) and (39), we obtain the ratio of the speed of sound in wide and narrow mounts in the following form:

$$\frac{c_1}{c_{bar}} = \sqrt{\frac{1-\mu}{(1-2\mu)(1+\mu)}}.$$

At the value $\mu = 0.49$ indicated above, the ratio is equal to

$$\frac{c_1}{c_{bar}} = \sqrt{\frac{1-0.49}{(1-0.98)(1+0.49)}} = 4.1.$$

The influence of a similar increase in the speed of sound in a broad mount upon its vibration insulation may be seen in the material of Chapter 12, in which vibration-isolating configurations are discussed.

In addition to longitudinal sound waves, propagation of transverse waves also may arise in solid media. Transverse vibration, and especially bending vibration may be of very great importance in the propagation of sound through plates and bars. These vibrations transmit a particularly large portion of the energy propagated through metallic structure of ships. The speed of bending waves in plates is equal to:

$$c_{bend.\ pl.} = \sqrt{2\pi f}\,\sqrt[4]{\frac{D}{M}} = \sqrt{2\pi f}\,\sqrt[4]{\frac{EI}{\rho h(1-\mu^2)}}. \qquad (40)$$

where $D = \frac{Eh^3}{12(1-\mu^2)}$ — stiffness in bending

$M = \rho h$ — mass per unit width of the plate, in grams;

$I = \frac{h^3}{12}$ — moment of inertia per unit width of the plate;

h — thickness of the plate, in cm;

μ — Poisson's ratio for the material of the plate.

The other factors have been introduced previously.

In the case of bars, we have similarly:

$$c_{bend.\ bar} = \sqrt{2\pi f}\,\sqrt[4]{\frac{EI_{bar}}{\rho S}} \approx 1{,}35\sqrt{c_l\, hf}\,,$$

where S is the cross-sectional area of the bar;

c_l is the speed of longitudinal waves;

$I_{bar} = \frac{bh^3}{12}$ is the moment of inertial of the cross-sectional area of the bar (b is the width of the bar).

The expression of the speed of propagation of bending vibrations may be simplified for particular materials. In the case of steel, for example, taking into account the relatively small value of μ, the speed of bending waves in bars and plates may be taken as equal, and may be determined according to the following formula:

$$c_{bend} \approx 950\sqrt{hf}\,. \tag{41}$$

The speed of propagation in formula (41), as in the other formulas, is expressed in cm/sec.

As is apparent, in distinction from longitudinal waves, the magnitude of the speed of propagation of bending waves depends upon the frequency, even for materials with low external friction (metals).

Example 6. The problem is to determine the speed of propagation of bending waves with frequency of 10 and 1,000 cycles in a steel plate 1 cm thick. What is the ratio of the value of this speed to the value of the speed of propagation of longitudinal waves?

Solution. We find the speed of propagation of the bending waves in the plate according to formula (41).

At a frequency of 10 cycles:

$$c_{bend} \approx 950\sqrt{1\cdot 10} \approx 3\cdot 10^3 \text{ cm/sec} = 30 \text{ m/sec.}$$

At a frequency of 1,000 cycles:

$$c_{bend} \approx 950\sqrt{1\cdot 1000} \approx 3\cdot 10^4 \text{ cm/sec} = 300 \text{ m/sec.}$$

The speed of propagation of bending waves at a frequency of 1,000 cycles is 10-fold greater than the speed of propagation of the same waves at a frequency of 10 cycles. It was mentioned earlier that the speed of longitudinal waves in metal was on the order of 5 km/sec. Consequently, the speed of propagation of longitudinal waves at a frequency of 10 cycles is approximately 170-fold greater than that of bending waves in a metal plate 1 cm thick, and 17-fold greater at a frequency of 1,000 cycles.

4. Sound Pressure, Acoustical Resistance, Intensity and Power of Sound

The displacement of particles of a medium during sound vibration may be expressed in the form

$$\xi = \xi_a \cdot e^{j(\omega t - kx)} \tag{42}$$

where ξ_a is the amplitude of vibrational displacement in the wave.

The exponent of the exponential function is distinguished from the corresponding exponent in the solution for a system with concentrated constants (formula 18) by the presence of a member depending upon the coordinate x, in addition to the presence of a time term. The former term indicates that the wave moves through space. The magnitude of k in this term is called the wave number.

With respect to finding the wave number we note that during the course of one period of vibration T the vibrating point moves through a distance equal to the length of the wave. From the expression of the indicator in the formula (42) it follows that

$$\omega T = k\lambda,$$

whence

$$k = \frac{2\pi}{\lambda}. \tag{43}$$

Taking account of the fact that according to formula (34):

$$\frac{\lambda}{T} = \lambda f = c,$$

the wave number k may be written in the following form:

$$k = \frac{\omega}{c}. \tag{44}$$

The speed of vibration of the particles of the medium is equal to:

$$\dot{\xi} = \frac{\partial \xi}{\partial t} = j\omega\xi_a e^{j(\omega t - kx)} = \dot{\xi}_a e^{j(\omega t - kx)}, \tag{45}$$

where

$$\dot{\xi}_a = j\omega\xi_a. \tag{46}$$

The speed of vibration of the particles of a medium must not be confused with the speed of propagation of vibration c. In ordinary sound processes the speed of vibration is thousands of times less than the speed of propagation of sound.

With the vibration of the particles of a medium an alternating pressure, called sound pressure, arises in the latter. Sound pressure is customarily expressed in dynes/cm^2, in newtons per m^2, or in "bars". It must be noted that the bar, as used in the USSR, is equivalent to the international "microbar".

For the purpose of determining the connection between sound pressure and other quantities characterizing the sound field, let us review the forces acting in a small layer of the medium, through which the sound wave is propagated.

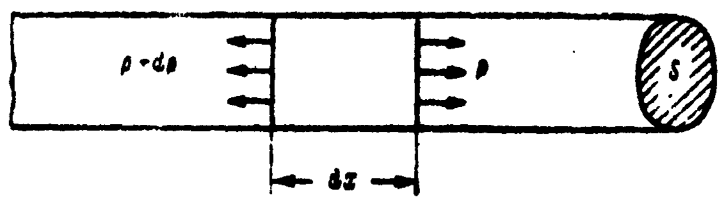


Figure 9. Determination of the connection between sound pressure and the speed of vibration in a plane wave.

The wave is described as plane, i.e. a wave in which the surface passing through a point with identical phase of oscillation is a plane, perpendicular to the direction of propagation of vibrations. In each element of this layer the elastic force must be balanced by the inertial force, i.e., we have the equation (in differentiated form):

$$S\,dp + \rho S\,dx \frac{\partial \dot{\xi}}{\partial t} = 0; \tag{47}$$

where dp is the change in vibratory pressure along the length of the element dx (Figure 9);

S is the cross-sectional area, perpendicular to the direction of propagation of vibrations;

ρ is the density of the medium; multiplying this value by c and by the length of the element dx, we obtain the mass of the elemental section of the medium;

$\frac{\partial \dot{\xi}}{\partial t}$ acceleration of vibration of the particles of the medium; the product of this acceleration and the mass of the medium element of area c and the length dx gives the inertial force of the element.

On the basis of formula (47) the vibratory pressure is equal to:

$$p = -\rho \int \frac{d\dot{\xi}}{dt}\, dx.$$

Inserting the value of the speed of vibration $\dot{\xi}$ from formula (45), we have:

$$p = -\rho \int \frac{\partial}{\partial t}\left(\dot{\xi}_a e^{j(\omega t - kx)}\right) dx = -j\rho\omega\dot{\xi}_a \int e^{j(\omega t - kx)}\, dx.$$

After integration the expression p acquires the form

$$p = \frac{\rho\omega}{k}\dot{\xi}_a e^{j(\omega t - kx)} + C = \rho c\dot{\xi} + C.$$

The integration constant c in the given case is equal to zero, because at $\dot{\xi} = 0$ there is no sound, and p = 0. We obtain

$$p = \rho c \dot{\xi}. \tag{48}$$

This equation is valid for any instantaneous values of speed of vibration and pressure, including amplitude values of these quantities:

$$p_a = \rho c \dot{\xi}_a. \tag{49}$$

The relationship obtained, in which the values of speed of vibration in the sound wave are connected with the alternating sound pressure in the waves, is one of the basic relationships in acoustics. It indicates that the sound pressure in the medium is directly proportionate

to the speed of vibration of the particles of the medium. The coefficient of proportionality is the value ρc, the product of the density of the medium and the speed of sound in it, and is called the specific acoustic resistance of the medium. For the sake of convenience it often is called simply the acoustic resistance, or the wave resistance of the medium, although strictly speaking the acoustic resistance, i.e., the resistance offered to the wave by the medium, is equal to:

$$s_0 = \rho c S, \tag{50}$$

where S is the area of the wave front.

If we inserted s_0 in formula (48) in place of ρc, we would have in the left portion of the equation not the vibratory pressure, but the force of vibration relative to the area S.

The name "acoustic resistance" is completely justified; from formula (48) it is apparent that at a given vibratory pressure, the speed of vibration of the particles of the medium is greater with a lower value of the acoustic resistance of the medium. The value of the acoustic resistance of several media (in a plane wave) is given in the last column of Table 1.

The expressions for the acoustic resistance of more complex cases of wave motion may be obtained in a similar manner, including that for a different kind of non-plane wave, and for transverse waves in solid bodies.

Acoustic resistance plays a primary role in the consideration of the phenomena of radiation, propagation, reflection and absorption of sound.

Example 7. A vibrating metal plate is bounded on one side by air, and on the other side by water. The problem is to evaluate the ratio of the magnitude of the sound pressure in the air and in water.

Solution. The speed of vibration of the particles of the medium on both sides of the plate is identical. From formula (48) it is apparent that in this case the ratio of sound pressure values in the two media is equal to the ratio of the acoustic resistance of the media, i.e.,

$$\frac{p_{water}}{p_{air}} = \frac{(\rho c)_{water}}{(\rho c)_{air}} = \frac{1.5 \cdot 10^5}{41} \cong 3560$$

It is apparent that at identical values of vibratory speed the sound pressure is thousands of times greater in water than in air. This results in the strong transmission of sound through water and fuel tanks in ships. Passing through the tank, the sound excites a vibration in the metal walls, which is accompanied by marked sound radiation in areas far removed from the machinery compartment.

Let us determine the value of the intensity, or force of sound. This value is taken to refer to the energy transmitted by a sound wave per unit time through a unit surface. The amount of work accomplished by sound pressure per unit surface is equal to:

$$dA = p\,d\xi = p\frac{d\xi}{dt}dt = \dot{\xi}p\,dt.$$

The work accomplished per period of vibration T is:

$$A = \int_0^T \dot{\xi}p\,dt.$$

The average value of this work gives the amount of energy transmitted through a unit of surface per unit of time:

$$J = \frac{1}{T}\int_0^T \dot{\xi}p\,dt.$$

Following insertion of the values ξ and p in the above expression in the form of sinusoidal functions of time and after integration we obtain an expression for the vector of the density of flow of energy (vector of Umov), or an expression of the intensity of sound in the form:

$$J = \frac{1}{2}\dot{\xi}_a p_a. \tag{51}$$

Inserting in the above the values $\dot{\xi}_a$ or p_a obtained from formula (49), expressed through wave resistance, the formula for J may be transformed to the following form:

$$J = \frac{1}{2}\frac{p_a^2}{\rho c} = \frac{1}{2}\dot{\xi}_a^2\rho c. \tag{52}$$

In practice, we often have to deal with not the amplitude values of speed of vibration and vibratory pressure, but the effective values of these quantities. The effective value of a variable parameter is taken as the value of that parameter which is equivalent to it in respect to energy and constant with respect to time.

It may be shown that the effective value of any given parameter described by a harmonic function of time is less than the amplitude of the value of this parameter by a factor $\sqrt{2}$. Thus for the effective values of the speed of vibration $\dot{\xi}_m$ and vibratory pressure p_m, we have:

$$\dot{\xi}_m = \frac{\dot{\xi}_a}{\sqrt{2}}; \quad p_m = \frac{p_a}{\sqrt{2}}. \tag{53}$$

Inserting the values of $\dot{\xi}_a$ and p_a from these expressions in formulas (51) and (52), we obtain the strength of the sound in the following form:

$$J = \dot{\xi}_m p_m = \frac{p_m^2}{\rho c} = \dot{\xi}_m^2 \rho c. \tag{54}$$

The sound power radiated by a source may be determined from the magnitude of the force of the sound. If the source radiates a planar wave, its power may be determined by multiplying the force of the sound by the cross-sectional area through which the sound energy is radiated. Most often, however, one encounters another type of radiation, the radiation of spherical waves. With this type of radiation, if the wave length is much greater than the dimensions of the radiator, the source is non-directional, i.e., it radiates sound in all directions.

At a sufficiently great distance from the source of a spherical wave, i.e., at a distance considerably greater than the wave length, the radius of curvature of the wave front is large and the wave approximates a plane wave, i.e., the above expression of sound pressure and strength of sound applies. The sound power radiated by the source then may be found from the expression:

$$W = S_c J = 4\pi r^2 J, \tag{55}$$

where S_c is the surface area of a sphere with radius r, equal to the distance from the source, at which the strength of the sound is equal to J.

Inserting the value of J from formula (54), we have:

$$W = 4\pi r^2 \frac{p_m^2}{\rho c} = 4\pi r^2 \dot{\xi}_m^2 \rho c. \tag{56}$$

Various cases may exist in which the simultaneous action of several sources of acoustic vibration is encountered. When vibration of the same frequency from two or more sources pile up exactly in phase or exactly out of phase the resulting vibratory pressure is determined by algebraic summation of the pressure of each of the sources.

More often cases are encountered in which the pressure of different sources have random phases. In the case of a large number of sources with random distribution of phases and frequencies, such as encountered in the case of noise, the resulting pressure is the energy summation of the sources, i.e., summation of their powers. The total power of an equivalent source W_e is equal to:

$$W_e = W_1 + W_2 + W_3 + \ldots = \sum W_i \tag{57}$$

where W_1, W_2, ... is the power of the individual sources.

The resulting sound pressure at a distance r from the sources is:

$$p_e = \frac{1}{r}\sqrt{\frac{\Sigma W_i \rho c}{4\pi}}. \tag{58}$$

Example 8. The sound pressure at a distance of one meter from a talking man is on the order of magnitude of 1 dyne/cm^2. The problem is to determine the acoustic power of an equivalent source radiating a spherical wave (let us assume, to the first approximation, that the sound of a voice conforms to this law).

Solution. From formula (56), we find:

$$W = \frac{4\pi (100)^2}{41} \approx 3 \cdot 10^3 \text{ ergs/sec} = 3 \cdot 10^{-4} \text{ watt.}$$

As is clear, the sound power of speech is very small. On the other hand, the sound power of noisy machinery can exceed tens and hundreds of watts.

Example 9. The sound pressure at a distance $r_1 = 5$m from the source of a spherical wave is $p_1 = 100$ dynes per cm^2. Determine the sound pressure p_2 at a distance $r_1 = 50$ m from the source.

Solution. From formula (56) it follows that with an unchanged power of the source the sound pressure is inversely proportional to the distance from the source. The pressure at a distance of 50 m is equal to:

$$p_2 = p_1 \frac{r_1}{r_2} = 100 \frac{5}{50} = 10 \text{ dynes/cm}^2. \qquad (59)$$

Example 10. With the operation of a non-directional point source (i.e. the source is of extremely small dimensions), radiating a spherical noise wave with power W, the sound pressure at a fixed distance is equal to p. What is the value of the sound pressure at this distance if the number of sound sources is increased by the factor m?

Solution. From formulas (56) and (58) it is apparent that the magnitude of the sound pressure of the noise is proportional to the square root of the power of the source. Utilizing these formulas, we obtain for the magnitude of the sound pressure, with the action of m sources:

$$p_m = p\sqrt{\frac{mW}{W}} = p\sqrt{m}. \qquad (60)$$

In particular, with the transition from one source to two sources the pressure increases approximately 40 percent, and with three sources the pressure increases 70 percent, etc.

5. Noise, Sound Spectra

With the exception of the last few formulas, the contents of the foregoing paragraphs relate mainly to simple sounds, or as they sometimes are called, pure tones, i.e., harmonic vibrations of a definite frequency and strength (force). In practice we often encounter complex sounds, constituting a mixture of several or many simple vibrations of various intensity and frequency.

A complex vibratory process of this type may be represented mathematically in the form of a summation of harmonic functions:

$$\gamma(f,\ t)=\sum_{i=0}^{\infty} A_i \sin(2\pi ift+\varphi_i), \qquad (61)$$

where A_i and φ_i are the corresponding amplitudes and phases of individual members of the series (i is any whole number);

f and t are frequency and time;

with a large i we have a Fourier series.

A series of the type of equation (61) may be constructed for the components of a vibratory pressure, for the frequency of vibration or any other parameters of a vibratory process.

From formula (61) it may be seen that the components of the series, forming a complex sound, may be included in either the function of time t, or in the function of frequency f. The depiction of a vibratory process or its individual components as a function of time is called an oscillographic process.

Examples of oscillograms of several vibratory processes are shown at the left in Figure 10. Figure 10 shows the oscillogram of sound pressure under simple harmonic vibration, in the absence of any other component except the components of main frequency. A similar oscillogram (for the amplitudes of vibratory displacement) is obtained directly in depicting the oscillation of a mass suspended from a spring (Figure 1).

Figure 10, b shows the oscillograms of two vibrations, each of which consists of two simple harmonic vibrations of different frequency. The frequency of the components of each vibration are in the ratio of 1 : 2, and the amplitudes in the ratio 1 : 1. The vibrations are distinguished from each other by the magnitude of phase displacement between their components. In one of the vibrations the phase displacement the components is equal to zero, and in the other is equal to $\frac{3}{2}\pi$.

Depiction of the human voice producing extended vowel sound produces a more complex oscillogram (Figure 10, b).

Oscillographic representation of vibratory processes is used often in structural mechanics in the investigation of low-frequency vibrations.

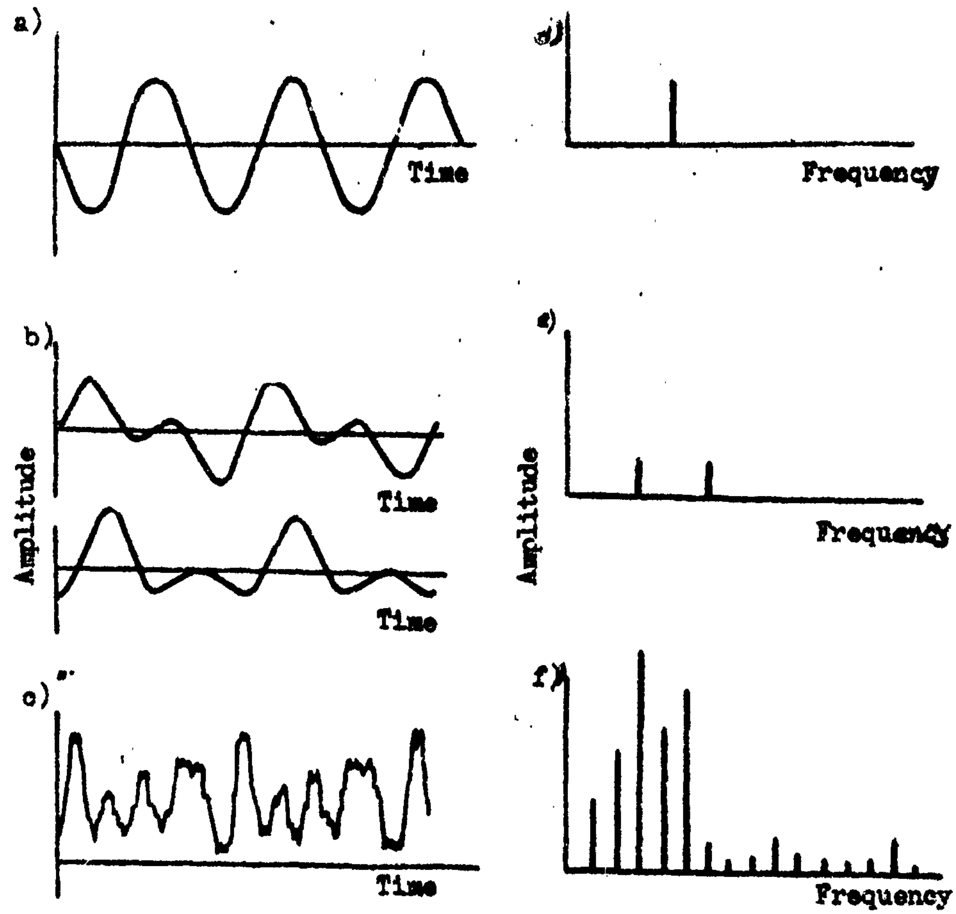


Figure 10. Oscillograms and spectrograms of several vibratory processes. Those on the left depict amplitude vs. time; those on the right, amplitude vs. frequency.

The disadvantage of this method is the well known cumbersomeness of the process of interpretation of the graphs, requiring special harmonic analysis. Because of this, another method of depiction of vibratory processes often is used in statistics, the depiction of their components as functions of frequency. This depiction is called the spectographic process.

Although a more complex apparatus is used in the formation of a spectrogram, the use of the latter greatly facilitates evaluation of the peculiarities of each complex vibration in relation to its aural perception. Aural perception is characterized by the fact that the ear is

not equally sensitive to sounds of different frequencies; on the other hand the spectrogram directly indicates the frequencies of the most intense components of a given sound, as well as the frequencies of of the less intense sounds.

The figures at the right in Figure 10 are spectrograms of the vibrations depicted by oscillograms at the left in Figure 10. The spectrogram in Figure 10, d depicts a sinusoidal vibration of fixed frequency. The frequency of the vibration is indicated on the horizontal axis, as in all the following spectrograms, and the amplitudes of the components are indicated on the vertical axis. Thus the amplitude of each of the components is determined by the height of the corresponding bar on the spectrogram. If a simple harmonic vibration is depicted by a single bar, vibrations consisting of two components are depicted by two bars (Figure 10, e).

Comparison of Figures 10, b and 10, e shows that the character of the spectrogram depends solely upon the value of the amplitude and the frequency of the individual component sounds, but not upon the magnitude of the phase displacement between them. This also corresponds to the properties of the ear, which reacts only to the magnitude of the amplitude and frequency of individual components of sound, independently of the phase relationships between them.

The spectrogram of Figure 10, f shows that the sound of the human voice has a large number of components of various frequencies and amplitudes.

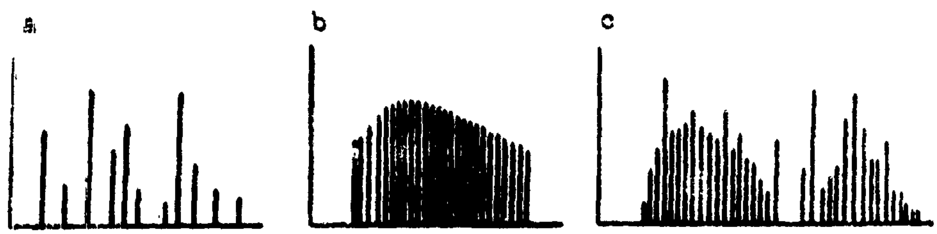


Figure 11. Several types of sound spectra.

Figure 11 shows various types of spectra of vibrations. A spectrum in which individual components are separated by more or less significant frequency intervals (Figure 11, a) are called, in analogy to the field of optics, a line or discrete spectum. This type includes, for example, the spectrum of the human voice discussed (Figure 10, f). The components of a line spectrum, the frequencies of which are close together, are called harmonics. The number and strength of the individual frequencies of the components of a sound determine its tone coloring, or timbre.

Figure 11, b is an example of another type of spectrum, the so-called continuous spectrum, in which the intervals between the frequency components are very small, i.e., the components follow closely after each other. A spectrum of this type arises within definite frequency ranges in the striking together of solid bodies, and in the formation of sound impulses. The shorter the impulse, the wider the band of frequencies in which the components of the sound are located.

The predominant type of spectrum in the acoustics of ships is the mixed spectrum (Figure 11, b). This type includes individual discrete components in the continuous portion of the spectrum. The majority of spectra of mechanical noises belong to this type. In electrical machinery having a marked magnetic sound the spectrum of vibrations may be of the line type.

A review of sound spectra leads us to a physical definition of the concept of noise. From a physical point of view noise may be defined as a mixture of sounds with frequencies and phases which are distributed irregularly. From the physiological point of view noise is characterized as a sound process which is more or less unpleasant upon perception, and constitutes a certain hindrance to work or rest.

If a noise is of a continuous spectrum and the amplitudes of all components within a broad range of frequencies are identical, in analogy to the corresponding term from the field of optics this noise is called a white noise.

In the case of a continuous spectrum within the frequency range of f_1 and f_2, the intensity of the noise is equal to:

$$J = \int_{f_1}^{f_2} J_1(f)\, df,$$

where $J_1(f)$ is the so-called spectral density of intensity, indicating the character of distribution of the latter along the frequency scale.

For "white" noise,

$$J_1(f) = \text{const} \quad \text{and} \quad J = J_1(f_2 - f_1),$$

i.e., the intensity is proportional to the width of the frequency band of the white noise.

CHAPTER 2

ELEMENTS OF PHYSIOLOGICAL ACOUSTICS

6. Perception of Sound. Decibels

The human hearing apparatus is a very well perfected and complex organ. It consists of three parts, the external, middle and inner ear. The external ear consists of the aural helix, performing the function of concentration of sound, and the external ear canal, which has a surface of approximately 0.5 cm^2. It is separated from the middle ear by the tympanic membrane, which is approximately 0.1 mm thick. Concerning the sensitivity of the ear it is sufficient to say that vibrations of the tympanic membrane on the order of magnitude of the size of an atom are perceived as sound. These vibrations are transmitted through a system of three small bones, called the hammer, anvil and stirrup, in the inner ear. Here the vibrations act on numerous nerve endings sensitive to sound, which are located within the auditory canal, each of which reacts to vibrations of a definite frequency.

Vibrations perceived by the ear as heard sound have a frequency of form 16 or 20 cycles to 16 or 20 kilocycles. This range is not identical for different people, and depends upon the age of the person and the condition of his hearing apparatus. With the extended action of intense noise the upper frequency limit of sensitivity of hearing may decrease 5 to 6 kilocycles.

Vibrations of a frequency below 16 or 20 cycles are not perceived by the ear as heard sounds, and are called infrasounds, and vibrations with frequency above 16 to 20 kilocycles, which also are not heard by the ear, are called ultrasounds.

The range of audibility of sounds is limited not only by definite frequencies, but also by definite values of sound pressure, or the strength of the sound. These limiting values of sound pressure are depicted by two curves in Figure 12. The lower curve corresponds to the threshold of audibility, or the threshold of hearing. In the range of frequency of 1,000 to 5,000 cycles the average value of threshold pressure for persons with good hearing approximates $2 \cdot 10^{-4}$ dynes/cm^2.

The sensitivity of the ear decreases in proportion to digression above or below this range, which is reflected in an increase in the value of sound pressure at which the ear begins to react to sound.

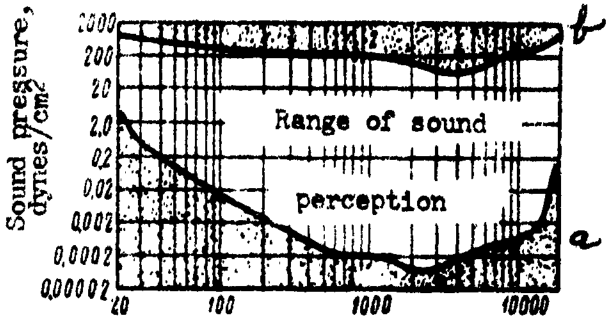


Figure 12. Range of sound perception in man.

(a) Threshold of hearing
(b) Threshold of pain

The upper curve in Figure 12 represents the threshold of pain. Sounds exceeding this threshold in magnitude may result in damage or destruction to the hearing apparatus.

At a frequency of 1,000 cycles, which is taken as the standard frequency for comparison in acoustics, the ratio of pressure at the threshold of pain to the pressure at the threshold of hearing is approximately 10^6, and the ratio of the corresponding intensities of sound attains a value of 10^{12}.

This large range of sound intensities perceived by the ear results from the fact that not the intensity of sound but its logarithmic is involved in sound perception. The corresponding logarithmic value β is called the level of intensity, or also the strength level of sound. Thus the level of intensity of sound is:

$$\beta = A \log J, \qquad (62)$$

where A is the coefficient of proportionality.

A graph of the logarithmic function is shown in Figure 13. At small values the argument of the slope of the logarithmic curve is great, and the slope decreases in proportion to the increase in the argument. According to this functional relationship the ear seems to increase its sensitivity in the perception of weak vibrations, and automatically "becomes calloused" in the perception of powerful sounds. This nonlinearity of the amplitudinal characteristics of sensitivity enables the ear to perceive an enormous range of sound pressures and intensities without overloading or distortion.

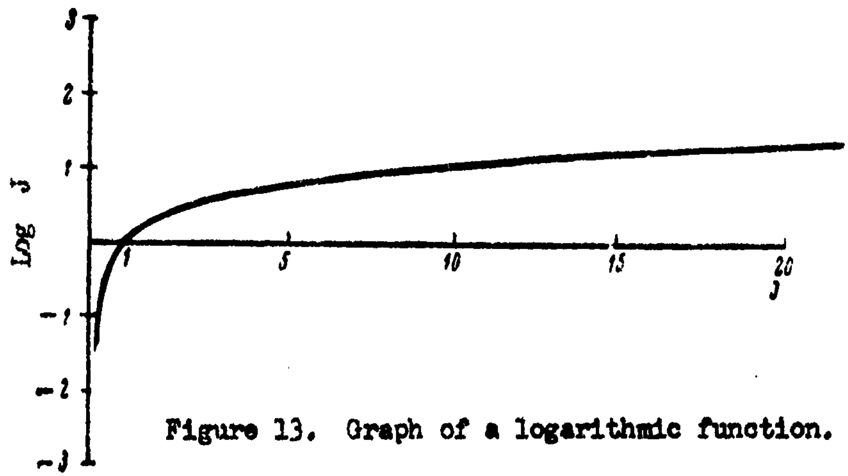


Figure 13. Graph of a logarithmic function.

Let us assume two sounds with intensities J_1 and J_2. According to formula (62) the level of intensity of these sounds is equal to:

$$\beta_1 = A \log J_1, \quad \beta_2 = A \log J_2.$$

The difference in the intensity levels of the two sounds is:

$$\Delta\beta = \beta_1 - \beta_2 = A\,(\log J_1 - \log J_2) = A \log \frac{J_1}{J_2}.$$

Assuming $A = 1$, we express the intensity level in bels.

It is more convenient to express β in decilogarithmic units, for which we must assume the value $A = 10$. In this case the difference between the two sound levels will be expressed in decibels. From a physiological point of view 1 db (decibel) corresponds to a hardly noticeable increase in the perception of the loudness of sound. The difference in the level of strength of two sounds, in decibels, is equal to:

$$\Delta\beta = 10 \log \frac{J_1}{J_2} \text{ db.} \tag{63}$$

Taking into account the relationship (54) between the intensity and pressure of sound, we obtain the following expression of the difference in intensity level of two sounds having pressures of p_1 and p_2 (the indexes m, indicating selection of <u>effective</u> pressure values, have been omitted here and in the following):

$$\Delta\beta = 10\lg\left(\frac{p_1^2}{p_2^2}\right) = 20\lg\left(\frac{p_1}{p_2}\right)\ \text{db.} \tag{64}$$

Table 2, which is suitable for practical computations, contains the ratios of sound pressures, sound intensity and their corresponding difference in sound strength levels, in decibels. If the value of this ratio is greater than unity it indicates a positive difference in level, and if less than unity it indicates a negative difference. A change in the sound level equal to 1 db (a value hardly detected by the ear) corresponds, as may be seen from the table, to a change in sound pressure of 12 percent, and a change of 26 percent in sound intensity.

Table 2 also may be utilized in calculating the difference in levels of sonic vibrations in decibels. For this purpose, the ratio of vibrational velocities must be used instead of the ratio of sound pressures. At fixed frequencies it may be replaced by the ratio of vibrational accelerations or displacements.

Finally, the table may be employed in computation of the amount of sound insulation and sound absorption, which also may be expressed in logarithmic units.

In acoustic calculations, as the zero decibel level it is customary to take a sound pressure $p_0 = 2 \cdot 10^{-4}$ dynes/cm^2, corresponding to the threshold of hearing at a frequency of 1,000 cycles. This pressure corresponds to a threshold of intensity of sound of:

$$J_0 = 10^{-9}\ \text{erg/sec cm}^2 = 10^{-16}\ \text{watts/cm}^2.$$

The strength of any given sound above the threshold is equal to:

$$\beta = 20 \log \left(\frac{p}{2 \cdot 10^{-4}}\right) = 10 \log \left(\frac{J}{10^{-16}}\right)\ \text{db}, \tag{65}$$

where p is the effective value of sound pressure in dynes/cm^2, and J is its intensity in watts/cm^2.

The expression "above threshold" is encountered very frequently, although it must be borne in mind that when we speak of the level of strength of a sound, its level is expressed with respect to the zero threshold.

TABLE 2

DECIBEL TABLE

Ratio of Sound Pressure $\frac{p_1}{p_2}$	Ratio of Sound Strength $\frac{\beta_1}{\beta_2}$	Difference in Sound Strength +db-	Ratio of Sound Pressure $\frac{p_1}{p_2}$	Ratio of Sound Strength $\frac{\beta_1}{\beta_2}$	Ratio of Sound Pressure $\frac{p_1}{p_2}$	Ratio of Sound Strength $\frac{\beta_1}{\beta_2}$	Difference in Sound Strength +db-	Ratio of Sound Pressure $\frac{p_1}{p_2}$	Ratio of Sound Strength $\frac{\beta_1}{\beta_2}$
1,0	1,0	0	1,0	1,0	11,2	126,0	21	0,089	$8\cdot10^{-3}$
1,12	1,26	1	0,89	0,79	12,6	158,0	22	0,079	$6\cdot10^{-3}$
1,26	1,58	2	0,79	0,63	14,1	200,0	23	0,071	$5\cdot10^{-3}$
1,41	2.00	3	0,71	0,50	15.8	251,0	24	0,063	$4\cdot10^{-3}$
1,58	2,51	4	0,63	0,40	17,8	316,0	25	0,056	$3,2\cdot10^{-3}$
1,78	3,16	5	0,56	0,32	20,0	398,0	26	0,050	$2,5\cdot10^{-3}$
2,00	3,98	6	0,50	0,25	22,4	501,0	27	0,045	$2,0\cdot10^{-3}$
2,24	5,01	7	0,45	0,20	25,1	631,0	28	0,040	$1,6\cdot10^{-3}$
2,51	6,31	8	0,40	0,16	28,2	794,0	29	0,035	$1,3\cdot10^{-3}$
2,82	7,94	9	0,355	0,13	31,6	10^{3}	30	0,032	$1\cdot10^{-3}$
3,16	10,00	10	0,32	0,10	100,0	10^{4}	40	0,010	10^{-4}
3,55	12,60	11	0,28	0,08	316,0	10^{5}	50	$3,2\cdot10^{-3}$	10^{-5}
3,98	15,8	12	0,25	0,06	10^{3}	10^{6}	60	$1\cdot10^{-3}$	10^{-6}
4,47	20,0	13	0,22	0,05	$3,16\cdot10^{3}$	10^{7}	70	$3,2\cdot10^{-4}$	10^{-7}
5,01	25,1	14	0,20	0,04	10^{4}	10^{8}	80	10^{-4}	10^{-8}
5,62	31,6	15	0,18	0,03	$3,16\cdot10^{4}$	10^{9}	90	$3,2\cdot10^{-5}$	10^{-9}
6,31	39,8	16	0,16	0,025	10^{5}	10^{10}	100	10^{-5}	10^{-10}
7,08	50,1	17	0,14	0,020	$3,16\cdot10^{5}$	10^{11}	110	$3,2\cdot10^{-6}$	10^{-11}
7,94	63,1	18	0,13	0,016	10^{6}	10^{12}	120	10^{-6}	10^{-12}
8,91	79,4	19	0,11	0,013	$3,16\cdot10^{6}$	10^{13}	130	$3,2\cdot10^{-7}$	10^{-13}
10,0	100,0	20	0,10	0,010	10^{7}	10^{14}	140	10^{-7}	10^{-14}

The value of the sound level also may be expressed through the value of the speed of vibration of $\dot{\xi}$ the particles of the medium, related to the sero threshold of speed $\dot{\xi}_0$.

$$\beta = 20 \log \frac{\dot{\xi}}{\dot{\xi}_0} \text{ db.} \qquad (66)$$

The expressions under the sign of the logarithm in formulas (65) and (66) are equal to each other. Taking into account the basic relationship between sound pressure and speed of vibration in a planar wave (formula 48), we obtain:

$$\frac{\quad}{2 \cdot 10^{-4}} = \frac{\dot{\xi}}{\dot{\xi}_0} = \frac{p}{\rho c\, \dot{\xi}_0}.$$

The value of the sero threshold of the speed of vibration may be obtained easily from the above. This threshold for air ($\rho c \approx 41$) is equal to:

$$\dot{\xi}_0 = \frac{2 \cdot 10^{-4}}{41} \cong 5 \cdot 10^{-6} \text{ cm/sec.} \qquad (67)$$

Example 11. Two machines are installed in a room; each of them has a sound strength level 81 db above threshold. With the aid of a series of measures the sound pressure of each machine is reduced to a value of 0.2 of the initial value. What will be the noise level in the room?

Solution. With the aid of Table 2 we find that reduction of the sound pressure to a value of 0.2 from the initial value corresponds to a change in level of approximately 14 db, i.e., the noise level of each of the machines is equal to 81 - 14 = 67 db.

From the same table we find that if the sound strength is doubled the sound level will be increased by 3 db. Thus the total noise level of the two machines after the measures have been taken for noise elimination is equal to 67 + 3 = 70 db.

It may be readily seen that direct computation consumes a great deal of time.

The noise level of each of the machines prior to elimination of noise is:

$$\beta_1 = 20 \log \frac{p}{p_0} = 81 \text{ db};$$

and after elimination of noise:

$$\beta_2 = 20 \log \left(0.2 \frac{p}{p_0}\right) = 20 \log \frac{p}{p_0} + 20 \log 0.2 = 81 - 20 \log 5 =$$
$$= 67 \text{ db}.$$

The noise level of the two machines is:

$$\beta_{total} = 67 + 10 \log 2 = 70 \text{ db}.$$

Example 12. Under the conditions of the previous example, determine the number of functioning machines necessary for raising the noise level in the room to 90 db.

Solution. From Table 2 we find that increasing the level by 20 db corresponds to a 100-fold increase in the sound strength, i.e., it is necessary to assume 100 sources of sound.

Let us determine the same case by direct computation.

The total level of n sources of noise is:

$$\beta_n = 10 \log \left(\frac{J_1}{J_0} \cdot n\right) = 10 \log \frac{J_1}{J_0} + 10 \log n \text{ db}.$$

According to the conditions of the example,

$$90 = 70 + 10 \log n,$$

whence $\log n = 2$, and $n = 100$.

This result convinces us of the improbability of obtaining large sound levels in rooms with the simultaneous functioning of several machines with low or medium noise levels.

Under the conditions of the above example the total noise level is not increased 20 db by the addition of a number of machines to bring the total number of machines to 100, because the addition of such a large number of machines would require very great enlargement of the room.

The graph of Figure 14 may be used for determining the total noise level of two sources with different noise levels. On this graph

the noise levels of both sources are depicted by straight lines, in which the noise level of the first source is represented by β db, and the level of the second source varies from $\beta + 10$ to $\beta - 10$ db. The total noise level is represented by a curve.

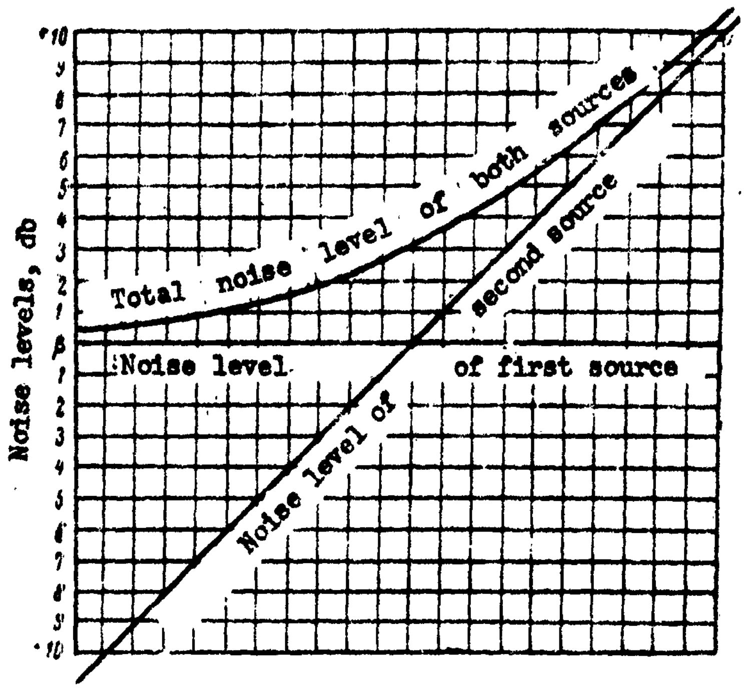


Figure 14. Total noise level of two sources with different noise levels.

At the point of equality of the noise level of both sources (point of intersection of the straight lines) we obtain the already familiar result: the total noise level is 3 db greater than the level of each of the sources. When the level of one of the sources is 6 db less than the level of the other source the total level is 1 db greater than that of the noisier source. If the difference between the levels of the individual sources exceeds 10 db the total noise level may be determined for practical purposes by the level of the noisier source.

Example 13. Five machines are located in a room of limited size. The noise level of three of the machines is 80 db, the level of the fourth is 83 db, and of the fifth is 77 db. The problem is to determine the value of the total noise level.

Solution. With a 3-fold increase (more precisely, 3.16-fold) in the intensity of sound, it may be seen from Table 2 that its strength increases by 5 db. Thus the total noise level of the first three machines is approximately 85 db. With the addition to this level of a value of 83 db the total level increases by 2.5 db (Figure 14) and attains a value of 87.5 db. The fifth mechanism does not add more than 0.5 db to this level. Consequently the total noise level of all five machines is equal to 88 db.

The logarithmic scale is valid not only for computing the perception of the strength of sound, but also of its frequency. As in perception of sounds of different strength, the ear notes changes not in the range of any given number of units (cycles), but in multiples. Because of this, for example, an increase in the frequency of sound from 100 to 200 cycles corresponds to the same perception of change in frequency in tone as in an increase in frequency from 1,000 to 2,000 cycles. This peculiarity of hearing is reflected in Figure 12 by the fact that the scale of frequency of the graph is plotted according to logarithmic functions.

The customary unit of reckoning intervals of frequency of sound vibrations is the octave, corresponding to a 2-fold change in frequency. Frequency ranges equal to half-octaves and one-third-octaves also are used extensively.

Sound vibrations may be perceived not only by the ear, but also directly through the bones of the skull. This type of perception of sound bears the name of bone conductivity. The level of air sound conducted by this means is 20 or 30 db lower than the level perceived by the ear. Bone conductivity plays a significant role in the perception of sonic vibration. Placing a steel bar against the bones of the head, with the other end resting on the housing of a machine, is sufficient for distinct detection of the timbre of the noise of the machine, the presence of impacting parts within the machine, changes in speed of revolution, etc.

7. Characteristics of the Loudness of Sounds of Different Frequencies

Evaluation of sound according to its strength above the threshold is valid only for a frequency of 1,000 cycles, corresponding to the

accepted standard threshold of pressure or strength of sound. At other frequencies the value of the zero threshold (Figure 12) changes, and the sensitivity of the ear to changes in the strength of sound also changes.

Figure 15 depicts a set of curves of equal loudness. Each of the curves characterizes a sound of differing frequency and intensity, although giving the impression of identical loudness. The bottom curve, as is the case with the analogous curve in Figure 12, corresponds to the threshold of perception of sound, and the top curve is the threshold of perception of pain.

The levels determined by the curves of Figure 15 are called levels of equal loudness. At a frequency of 1,000 cycles the levels of equal loudness are taken as equal to the corresponding levels of the strength of the sound; at other frequencies they differ to greater or lesser degree from the levels of the strength of sound. The level of the loudness of a sound of any given frequency is the level of strength of a sound with frequency of 1,000 cycles, of equal loudness to that of the given sound.

The loudness level is expressed in special units, phons. If, for example, a sound of any particular frequency produces an impression of being identical in loudness with a 1,000-cycle tone, the strength level of which is 50 db above threshold, it means that the level of loudness of the former is equal to 50 phons.

In the range of frequencies below 500 cycles and intensity level values below 60 db the steepness and the nearness of the curves of equal loudness are greatest. The latter means that in this range the ear is most sensitive to changes in the frequency and strength of sound. However, the absolute sensitivity of hearing in this range is low, taken per se (the threshold of hearing is 20 to 60 db higher than at a frequency of 1,000 cycles).

At values of the strength of sound greater than 80 db the curves of equal loudness are almost parallel to the horizontal axis, i.e., the values of the loudness level at any given frequency may be considered approximately equal to the corresponding values of the level of the strength of the sound.

Example 14. Determine the level of loudness of two sounds: the frequency of the first is 40 cycles, with a pressure of 2 dynes/cm^2, and the second sound has a frequency of 600 cycles and a pressure of 0.2 dynes/cm^2.

Solution. From Figure 15 we find that a curve of equal loudness corresponding to 60 phons passes through the point with coordinates of 40 cycles and 2 dynes/cm^2. Therefore the loudness level of the first sound is equal to 60 phons. The loudness level of the second sound also is equal to 60 phons, because the very same curve for 600 cycles frequency has an ordinate of 0.2 dynes/cm^2.

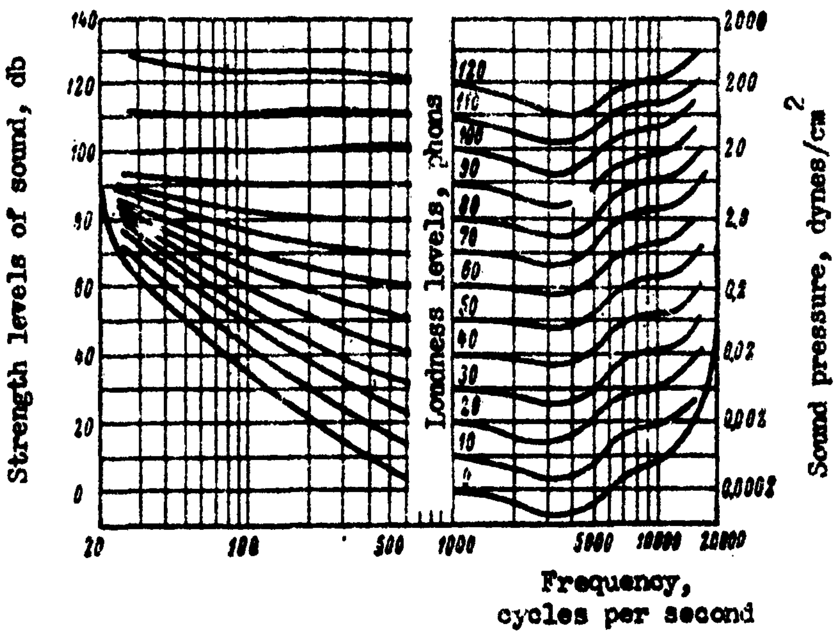


Figure 15. Curves of equal loudness of sounds.

Example 15. The frequency of the main component of the noise of a transformer, determining the total level of noise, is 100 cycles and it has a loudness of 100 phons. To what extent must the noise be reduced to bring its loudness level below 60 phons?

Solution. From the curve of equal loudness we find that at a frequency of 100 cycles a loudness level of 100 phons corresponds to a sound strength level of 100 db, and a loudness level of 60 phons corresponds to a strength level of 71 db. Therefore the strength of the sound must be reduced by 100 - 71 db = 29 db. As may be seen from Table 2, this corresponds to a 28-fold decrease in sound pressure, and a reduction of approximately 800-fold in the sound intensity.

Instruments used for the measurement of the loudness characteristics of sound (sound- or noise-meters) are unable to reproduce all curves of equal loudness. Figure 35 contains curves of equal loudness, plotted with an interval of 10 phons, although as mentioned in the foregoing, the ear detects changes in the strength of sound equal to 1 db, and because of this the actual number of curves of equal loudness is much greater. The approximate value of the loudness level obtained through the measurements of objective instruments is called the sound level.

This quantity is accepted in the acoustic standards of several countries. The sound level is expressed in decibels. At strength levels above 70 to 80 db the sound level, and the loudness level, also, coincide with the level of strength of the sound.

Table 3 includes values of the sound level for several frequently encountered sound and noise sources (similar levels for shipboard noise sources are presented in Chapter 5).

TABLE 3

SEVERAL SOUND LEVELS

Character and Sources of Sounds, Thresholds of Hearing	Sound level relative to $2 \cdot 10^{-4}$ dynes/cm^2
Threshold of hearing	0 - 10
Rustling of leaves, weak wind	10 - 20
Whisper, at distance of 1 m	30 - 40
Very soft music (radio)	40 - 50
Noise in room with window on street	40 - 50
Soft speech	50 - 60
Loud speech at distance of several meters	60 - 70
Music (through loudspeaker)	70 - 80
Street noise	70 - 80
Noise in factory shop	90 - 100
Orchestra music (fortissimo)	100 - 110
Noise of pneumatic tool	110 - 120
Aircraft propeller at 3 m distance	120 - 130
Threshold of pain	120 - 130
Noise of aircraft jet engine at 1 m from exhaust (to the side)	130 - 140
Noise of rocket installation at 20 meters	> 130 - 140

The above sound levels do not exhaust attempts at establishing the physiological characteristics of sound. The attempts at establishing them also were aimed at determining the characteristics of sound which would directly indicate the multiple by which one sound is louder than another. As a result of many investigations, of which we mention the very basic works of G. Fletcher (US), a loudness scale was established, i.e., values which are directly proportional to the perception of the loudness of sound of any given frequency.

A special unit, the son, was proposed for expression of the numerical value of loudness. The relationship between the loudness number of a sound in sons and its loudness level in phons (or the sound level in db) is shown in the form of a nomogram (Figure 16). The loudness value of 1 son corresponds to the loudness level of 40 phons. For loudness levels greater than 40 phons a change in the loudness level of a sound equal to 9 or 10 phons corresponds to a two-fold change in the perception of the loudness of the sound.

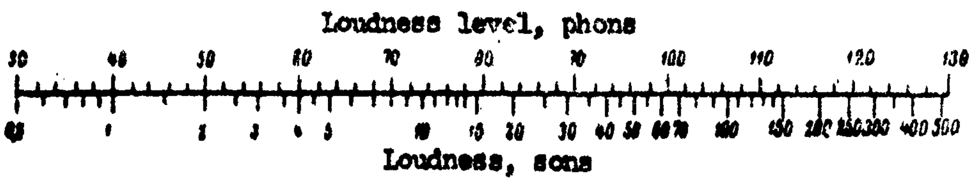


Figure 16. Relationship between loudness level and loudness of a sound.

With a change in the loudness level from 30 to 130 phons the loudness number (sons) changes more than 500-fold. However, a 2- to 4-fold change in the amount of loudness may be discerned by hearing; other relationships of loudness have not been subjected to any type of evaluation. Because of this, only limited portions of the nomogram are utilized in comparing the loudness of two sounds, corresponding to a change in loudness level of more than 20 to 25 db. The distribution of sections on the nomogram is determined by the magnitude of the loudness levels of the compared sounds.

The loudness scale also is utilized in calculation of the loudness levels of complex sounds. Utilization for this purpose is based on the fact that the loudness number is directly proportional to the perception of loudness, and because of this the loudness of a complex sound is equal to the sum of the loudness of its individual frequency components.

Example 16. Analysis of a ship's whistle and the sound of a rotary air blower indicated the presence of many tonal components in the spectrum of these sounds, determining the total sound level. The strength levels of these components at a distance of 5 meters are indicated together with their frequency in the corresponding columns of the table below. The problem is to determine which of the sounds is louder.

Solution. We determine the loudness level of the individual components L_i according to the equal loudness curves, and then their loudness G_i is determined according to the nomogram of Figure 16. The total loudness is found by summation of the loudness of the individual components. In the general case, in computing the loudness of a complex sound the mutual effects of the components in the course of their perception by hearing must be taken into account (masking effect). With sufficiently large frequency intervals between the components their mutual masking effect may be ignored.

Using the total loudness values thus found, the nomogram (Figure 16) again is used in determining the level of loudness of the sound of the whistle and the blower. The computation is shown in the following table.

Sound of Whistle				Sound of Blower			
Given		Found		Given		Found	
Frequency f_i, cycles	Intensity of sound β_i, db	From equal loudness curve — Loudness level of sound, L_i, phons	From nomogram (Figure 16) — Loudness G_i, sons	Frequency f_i, cycles	Intensity of sound β_i, db	From equal loudness curve — Loudness level of sound, L_i, phons	From nomogram (Figure 16) — Loudness G_i, sons
100	90	90	32	800	80	80	16
250	110	108	112	1400	70	70	8
400	90	90	32	3200	80	85	23
1200	85	85	23	4500	75	75	11
				7000	100	95	46
		ΣG_i	199			ΣG_i	104
According to Nomogram, Figure 16, L_e = 116 phons				According to Nomogram, Figure 16, L_e = 107 phons			

The loudness level of the sound of the whistle is 9 phons greater than the loudness level of the sound of the blower, i.e. the sound of the whistle is approximately twice as loud as the sound of the blower.

8. Irritating Effect of Sounds

Perception of the loudness of sound does not coincide with the irritating effect of sound. The problem of the irritating and fatiguing effects of sound were investiged for the first time in the decade of 1930 by Laird and Coy [approximate transliteration from the Cyrillic], who constructed the so-called curves of equal unpleasantness of sound on the basis of statistical experiments.

During recent years this problem has received further precise investigation and development. Figure 17 shows curves of equal unpleasantness of sounds, obtained by D. Parkinson (157). The loudness level of the sound in phons is plotted on the vertical axis of the graph; the ordinates of the graphs indicate levels of unpleasantness of sounds of different frequency.

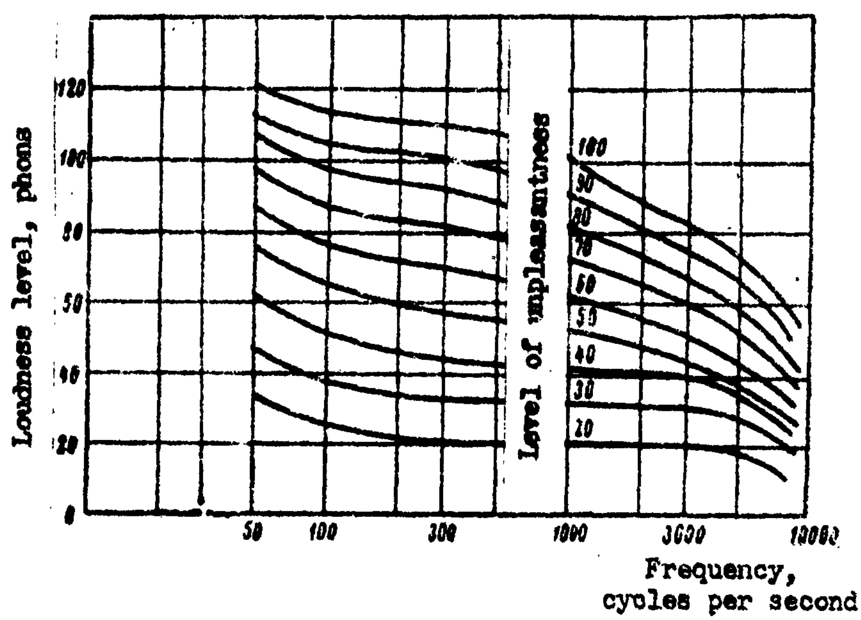


Figure 17. Curves of equal unpleasantness of sounds.

No special units have been proposed for the level of unpleasantness. Some authors have suggested expressing the level of unpleasantness in decibels.

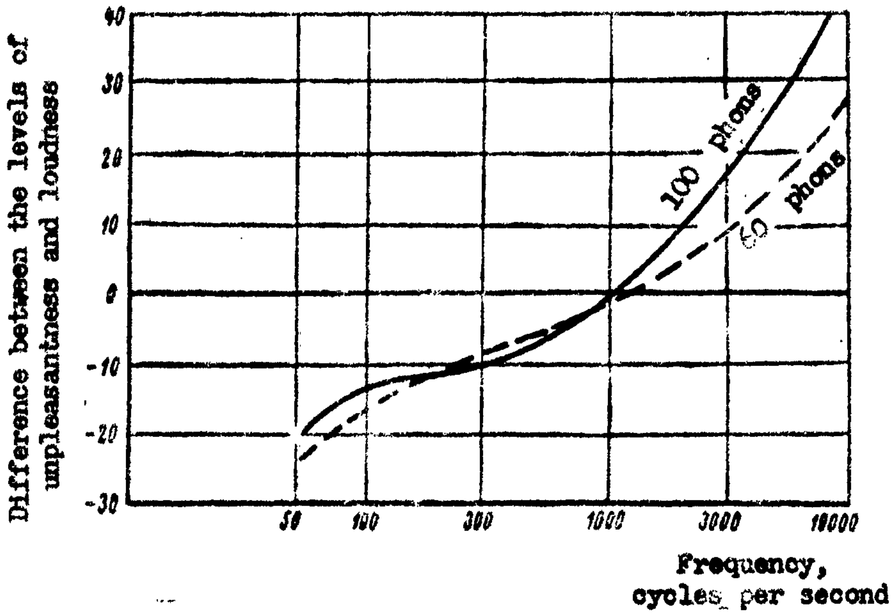


Figure 18. The surpassing of the loudness level of a sound by its level of unpleasantness (according to data of Figure 17).

The graph of Figure 18 was constructed by the present author on the basis of the data of Figure 17. It reflects the surpassing of the loudness level by the level of unpleasantness of sounds. The solid curve corresponds to a very loud sound (loudness level 100 phons), and the broken curve corresponds to the sound of average loudness (60 phons). Although the graph is not rigorous because it was obtained through a computation using different units, it suffices to show fairly clearly the difference in the loudness and the unpleasant effect of sounds. It is apparent that at high frequencies the perception of unpleasantness of sound is 20 to 30 logarithmic units above the perception of loudness. On the other hand, at low frequencies the value of the loudness level surpasses the values of the level of unpleasantness.

Example 17. Under the conditions of the previous example, let us evaluate which of the sounds, the sound of the whistle or the sound of the rotary air blower has the more irritating effect.

Solution. We shall attempt to perform the required evaluation using the most intense component of each sound. The most intense component (108 phons) in the sound of the whistle has a frequency of 250 cycles. At this frequency, as may be seen from Figure 17, the loudness level surpasses the level of unpleasantness by 10 units.

At a frequency of 7,000 cycles, where the strongest component of the sound of the blower is located, the level of unpleasantness surpasses the loudness level by approximately 40 units. At this level of loudness this component is 13db below the strongest component of the sound of the whistle. Taking all these values into account, we find that the excess of the level of unpleasantness of the sound of the blower above the corresponding level of the sound of the whistle is -10 + 40 - - 13 = 17 units.

Therefore the high-frequency sound of the blower, which was of a lower level, was found to be considerably more unpleasant than the low-frequency sound of the whistle. Because of this it is often sufficient to reduce the intensity of the high-frequency component of the noise to effect a marked reduction in its irritating effect, although this does not change the total level of noise.

The presence of a large number of components with high frequency in a noise is one of the factors determining the considerably irritating effect of the noise. Another factor of this type is the intermittent character of the noise. The ear accomodates itself to noise of a given level, and changes in this level are perceived as being very unpleasant. A periodically operating hand pump located near a passenger cabin may have a more irritating effect than the regular, though somewhat louder noise transmitted from the machinery compartment.

The loud speech of nearby persons, cries of children and loud radio playing have a considerably irritating effect on a resting person. Even though the meaning of speech or of the exclamations are not distinguishable, but their intonations are perceivable, the psychophysiological effect upon a resting man or person occupied in intellectual work is rather strong.

The result of the effect of noise on the organism is not limited to irritation. Investigations have shown that the speed of visual reaction is reduced 25 percent in the presence of a noise of 90 phons, and the time required for resolution of typical task tests increases

10 percent above average (154). Noise acts upon muscular activity, and may lead to a drop in the pulse (down to 70 percent of normal) as a result of an increase in resistance to the circulation of the blood.

As is well known, periodic relaxation of the nervous system is necessary for productive work of man. Under noisy conditions the necessary relaxation does not occur. Numerous investigations in the USSR indicate that in this case the decrease in work productivity mentioned above, and premature deterioration of the organism, result. At high noise levels (110 to 120 db) not only the productivity of any given work of a person is reduced, but the possibility of damage to his nervous system and hearing apparatus exists.

The facts stated in the present chapter concerning the levels, and physical and physiological characteristics of sound are summarized in Table 4.

TABLE 4

CHARACTERISTICS OF SOUND (NOISE)

Characteristic	Principal Function of the Characteristic	Units
Level of Strength of Sound (Intensity)	General evaluation of the strength of sound, comparison of the strength of sounds	Decibel
Level of Loudness of Sound	Determination of the level of auditory perception of sounds of various frequencies, and complex sounds	Phons
Sound Level	Approximate expression of the level of loudness of sounds in measurement by objective noisemeters	Decibel
Loudness of Sound	Computation of the loudness of complex sounds, direct comparative evaluation of the loudness of sounds differing several-fold in loudness	Sons
Level of Unpleasantness of Sound	Comparative evaluation of the unpleasant effect of sounds of different frequencies and strength	--

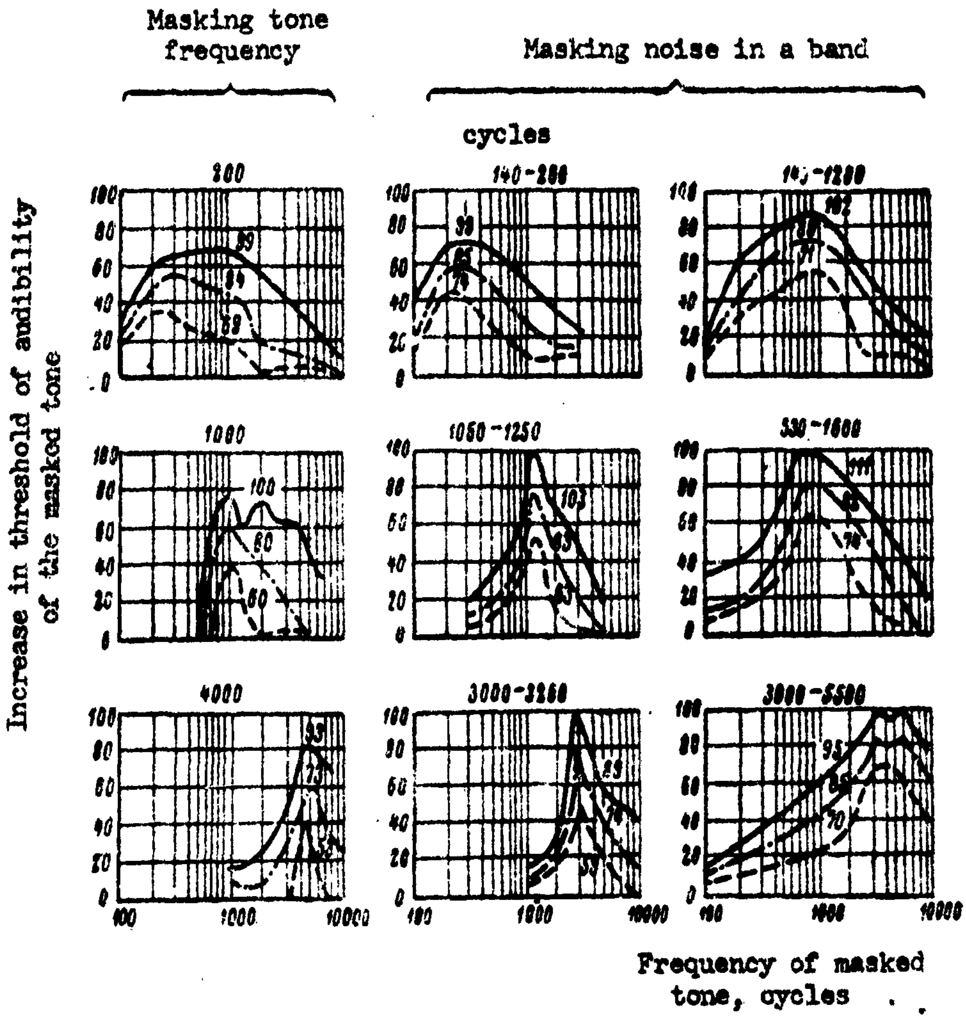


Figure 19. Masking effect of pure tones and noises (in a band) upon pure tones of various frequency.

All the levels listed in the table are utilized to greater or lesser degree in acoustic investigations and computations. In the USSR the level of intensity of sound, and the level of loudness of sound, plus units for their measurement are standardized (GOST 8849-58, 1 January 1959).

9. Masking Effect of Sounds

Reduction in the ability of a listener to perceive one sound in the presence of another sound is called masking of sound. In this, the first is called the masked, and the second the masking sound.

At the present time the problem of the masking of sounds has been sufficiently well studied. Research has been conducted on the masking effect of pure tones on other pure tons and of noises on pure tones, as well as the masking effect of noises or impulse-tones on human speech.

Figure 19 contains curves of the masking of pure tones by tones of 200, 1,000 and 4,000 cycles (left column) frequency, and by noises having relatively narrow (center column) and broad (right column) ranges of frequency obtained by Fletcher. The masking effect appears in the increase in the threshold of audibility of the masked tone (i.e., in a reduction in the sensitivity of hearing a tone of this frequency), in the presence of a masking sound. At low frequencies the masking sound (including high levels of the latter) the masking effect extends to a broad range of frequencies, which mainly are higher than the frequency of the masking tone. With an increase in the frequency of the tone of masking, the range of masking frequencies becomes narrower. In other words, the greatest masking effect is exhibited by sounds of low frequency. Because of this the low-frequency sound of a ship's whistle, discussed in Example 16, has a very strong masking effect.

It may be seen from Figure 19 also, that the masking effect of noise is somewhat higher than the masking effect of pure tones.

Figure 20 illustrates the masking effect of noise upon human speech. The lower curve is an idealized spectral curve of speech (loud voice). The broken curve corresponds to the threshold of masking of speech, i.e., the level of noise which renders speech completely unintelligible. The shaded area determines the frequency in the spectrum of speech which determines its intelligibility to the greatest extent. This area may be masked only by powerful sounds of low frequency, or frequencies coinciding with the given range. Sounds with 4,000 cycles or higher frequency have a weaker masking effect upon this area. In this way we arrive at the interesting conclusion that high frequency

ounds of average strength are fairly unpleasant in respect to effect upon the organs of hearing and upon the nervous system, although they do not mask the human voice.

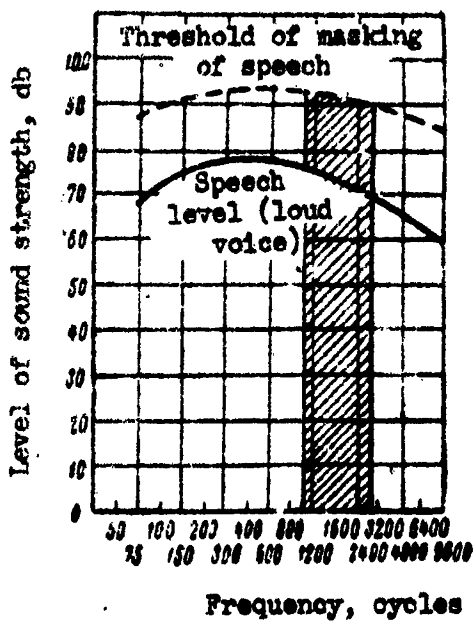


Figure 20. Spectral composition of speech, and the threshold of its masking.

For quantitative evaluation of the degree of intelligibility of speech under conditions of interference the articulation tables given in reference (28) are used. They consist of one hundred words, mostly highly articulated, and therefore very difficult to understand. The number of words correctly understood determines the percentage of verbal articulation. Under the conditions of the conduct of shipboard work, 40 to 50 percent of the articulation may be recognized sufficiently, although under the same conditions the number of sentences understood correctly exceeds 90 percent. With a verbal articulation of 70 percent the percentage of understood sentences is approximately 100, and the sense of sentences is understood almost without effort.

Figure 21 shows the relationship between verbal articulation of speech, the speech level and the noise level. In the given case the noise covers a broad range of frequencies, from 125 to 5,700 cycles. In this range noise has a continuous spectrum of constant amplitude, i.e., it is a "white" noise. In the experiment the noise level is varied from 0 to 120 db. The speech level also changes within a broad range, up to the levels corresponding to a loud cry (on the order of 90 db) or loud radio transmission of speech (90 to 100 db). The solid curves correspond to the values of articulation at the given corresponding speech level and level of interfering noise, and the broken curves unite the points of the basic curves containing a constant difference between the speech level and noise level.

Example 18. The noise level at the control station of the machinery compartment of a ship is 90 db; the character of the noise approximates that of a "white" noise. The problem is to determine the magnitude of the speech level necessary for attaining 50-percent articulation. What is the amount of decrease in articulation at this level of speech if the level of the interfering noise is increased to 110 db?

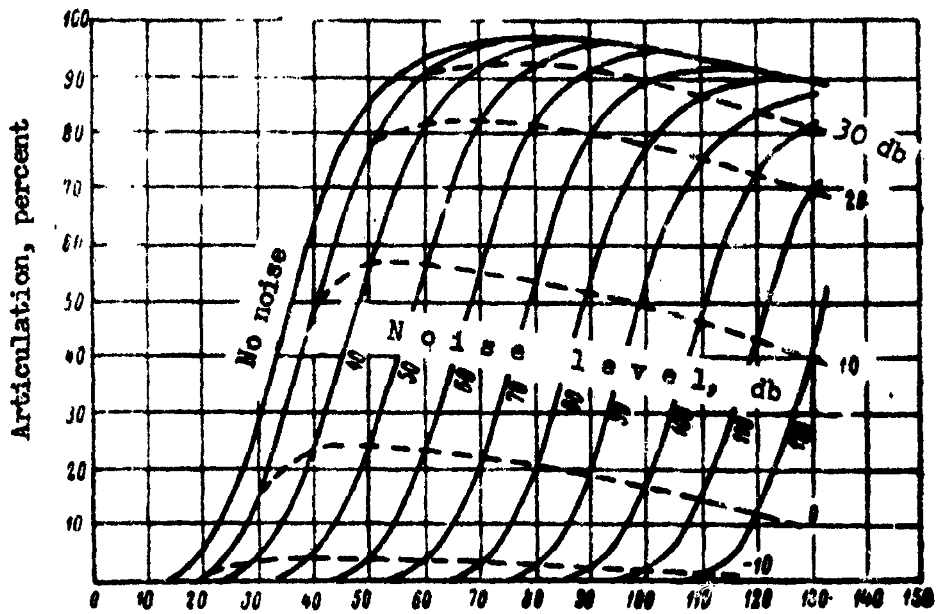


Figure 21. Dependence of the articulation of speech upon the level of speech and the level of the interfering "white" noise.

------- Difference between the speech and noise level (db)

Solution. According to the curves of Figure 21 we find that at a noise level of 90 db a speech level no greater than 100 db is necessary for obtaining 50-percent articulation, i.e., the difference between the speech and noise levels is + 10 db.

If the noise level is increased to 110 db this difference is equal to - 10 db. Then we find the point of intersection of the broken curve corresponding to -10 db with the abscissa of 100 db. The articulation found is less than 5 percent, i.e., intelligibility of speech drops very sharply.

At the same difference between the noise and speech levels the articulation increases somewhat, as the noise level decreases to 50 db, which may be seen from the course of the broken curves in Figure 21. When the noise level decreases from 100 to 50 db (and with a corresponding reduction in the speech level) the articulation attains a relative increase of 7 percent.

10. Perception of Vibration

Vibration not only is a source of airborne noise, but in itself may produce an unpleasant physiological effect. One of the first investigations of the effect of vibration upon the human organism was performed by the Soviet scientist Ye. Ts. Andreyeva-Galanina (9).

Most investigators have studied the effect of artificially generated vibration on individual parts of the human body. An article by William (169) contains data on investigation of the influence of ship vibrations on the human organism.

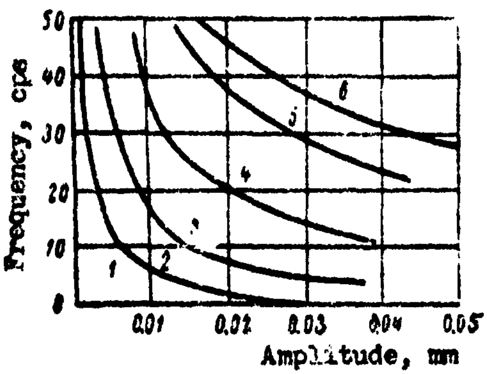


Figure 22. Sensitivity of man to shipboard vibrations.

1 - Vibration not perceived
2 - Vibration weakly perceived
3 - Vibration markedly perceived
4 - Vibration unpleasant
5 - Vibration markedly discomforting
6 - Vibration causes pain

The author divides vibrations into six degrees, or gradations, according to the degree of physiological perception of the vibration. The ranges corresponding to these gradations of vibrations are shown in Figure 22. They include vibrations with frequency up to 50 cycles and amplitude up to 0.05 mm (a total movement of 0.1 mm). From the curves it may be seen, for example, that at a frequency of 20 cycles unpleasant perception begins, even when the amplitude is only 0.015 to 0.02 mm. Human sensitivity increases sharply with an increase in the frequency of vibration.

Vibration, like sound, customarily is expressed in logarithmic units above a threshold. If a vibration of 0.001 cm/sec is taken as the threshold, the level of vibration will be expressed in pals [transliteration; "pal" = pawl]. A level of 60 pals comprises a vibration which is very unpleasant to man.

M. S. Antsyferov (12) suggested the expression of the level of the speed of oscillation of a vibration above a threshold equal to $5 \cdot 10^{-6}$ cm/sec. This value, as was seen from formula (67), is nothing other than the speed of oscillation of air particles at the threshold of aural perception. The level of vibration above the indicated threshold is equal to:

$$\beta_b = 20 \log \left(\frac{\dot{y}}{5 \cdot 10^{-6}} \right) \text{ db} \qquad (68)$$

where $\dot{y}$ is the speed of vibration of the vibrating surface, in cm/sec.

The adoption, for the zero level of vibration, of the vibratory velocity of air particles at the threshold of aural perception, is convenient because it serves to relate very closely the vibrations of a surface and the sound radiated by it.

CHAPTER 3

SOME SOUND PROCESSES

11. Radiation of Sound

The phenomenon of radiation of sound during the operation of machines and in the vibration of the structural boundaries of a compartment has a fairly complex character. In many cases these complex actual processes and sources of radiation may be reduced to simplified models. This facilitates analysis of the relationships between the parameters of the process of radiation and the parameters of a source of vibration, and enables identification of the most rational way in which sound radiation may be reduced.

Figure 23 contains diagrams of some extremely simple sources of vibration and depicts the radiation of sound by the action of these sources. Figure 23 a shows the radiation of a rigid, very long wall or panel undergoing synphasic vibration (i.e. vibrations with identical phase for the entire surface of the panel) and producing these vibrations in a direction perpendicular to its surface. The term "very long" is taken to mean that the dimensions of the panel exceed many-fold the wave length λ in the medium surrounding the panel. Somewhat similar to this process may be the vibrations of thick metallic walls (for example, the flat walls of the cylinder block of an engine) at high sonic frequencies, at which the wave length of the sound in air is small.

The process depicted in Figure 23-a is a unique type of radiation, in which the sound pressure in the medium is directly proportional to the speed of vibration, independently of frequency. The wave in the medium has a flat front, and the amplitude of the sound pressure is equal to (formula 49):

$$p_a = \rho c \dot{\xi}_a .$$

Because there is no lateral displacement of the medium, the speed of vibration of the particles of the medium $\dot{\xi}_a$ may be replaced by the speed of vibration of the surface of the radiator $\dot{y}_a$. Converting

to the average value of the speed of vibration $\dot{y}$, we obtain the level of sound pressure above the threshold:

$$\beta = 20 \log \frac{\rho c \dot{y}}{2 \cdot 10^{-4}} = 20 \log \frac{41}{2 \cdot 10^{-4}} \dot{y} = 20 \log \dot{y} + 106.5 \text{ db.} \tag{69}$$

This is the expression of the sound level when the absolute magnitude of the speed of vibration of the surface is known. If this vibration is expressed with respect to the level of the vibrational velocity above a threshold of $5 \cdot 10^{-6}$ cm/sec, it may be seen that this level also is equal to the sound level in the medium. Actually, the sound level is equal to (formulas (66) and (67)):

$$\beta = 20 \log \frac{\dot{\xi}}{5 \cdot 10^{-6}} \text{ db}$$

The level of vibration (68) is:

$$\beta_b = 20 \log \frac{\dot{y}}{5 \cdot 10^{-6}} \text{ db.}$$

However, because $\dot{\xi} = \dot{y}$, we have

$$\beta = \beta_b. \tag{70}$$

The flat surface of a panel with synphasic vibration is the most efficient of all possible sound radiators. Thus by measuring the level of the speed of vibration of any given long panel we obtain the maximum possible sound level in the medium surrounding the panel. In this, it is natural that changing the level of vibration by a certain number of decibels evokes the same change in sound level of the medium.

A point source is a less efficient radiator of sound. A source of this type may comprise a sphere pulsating in synphase, the radius of which sphere is considerably less than the wave length of the radiated sound (Figure 23-b). In this case the amplitude of sound pressure at a distance r is many-fold greater than the wave length of the sound, and is equal to:

$$p_a = \frac{f \rho}{2r} Q_a \tag{71}$$

where f is the frequency of vibration;

ρ is the density of the medium;

Q_a is the amplitude of the volumetric speed of the surface of the source, or the so-called output of the source, equal to the product of vibratory speed of the surface of the source by its area.

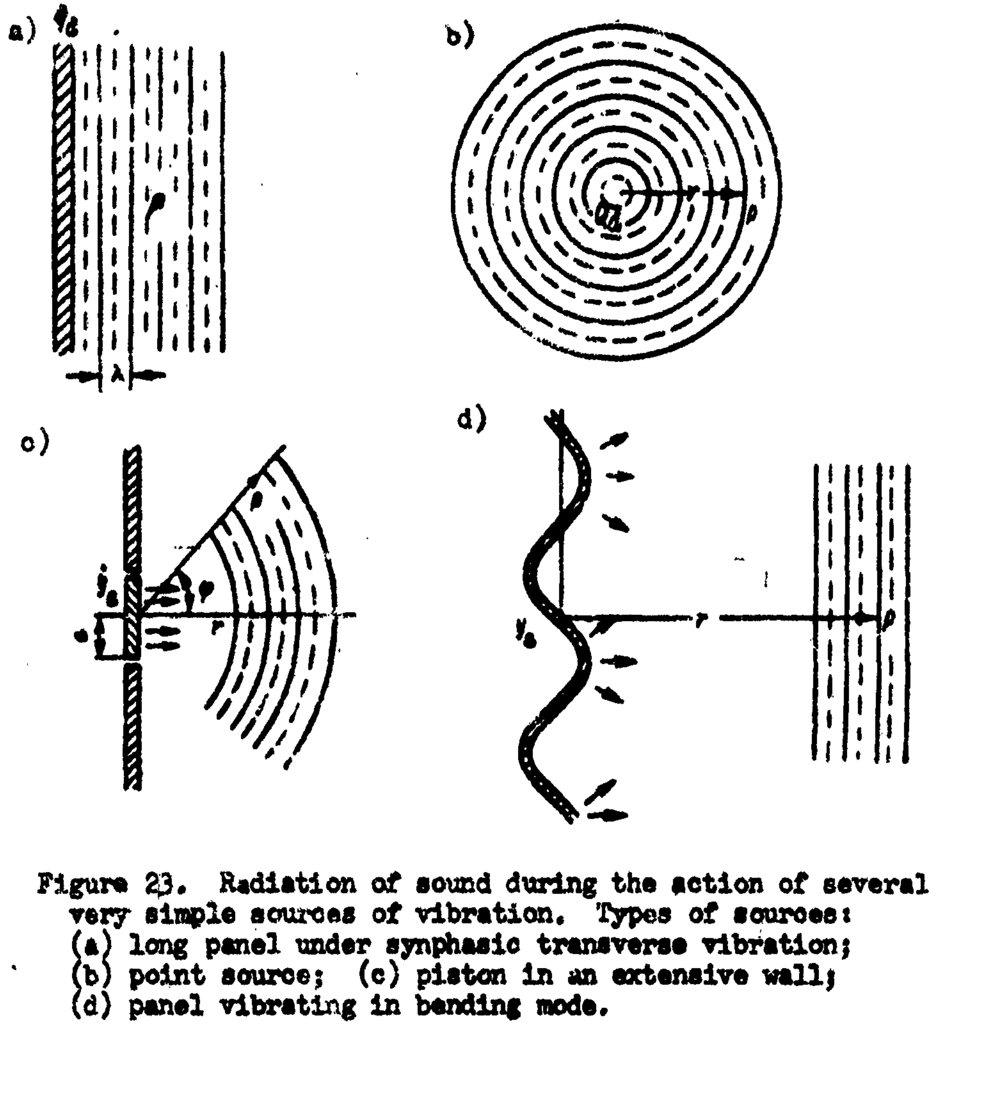


Figure 23. Radiation of sound during the action of several very simple sources of vibration. Types of sources: (a) long panel under synphasic transverse vibration; (b) point source; (c) piston in an extensive wall; (d) panel vibrating in bending mode.

It is apparent that the sound pressure in this case is inversely proportionate to the distance. This law of measurement of pressure was mentioned earlier (cf., for example, formula 58); it corresponds to the divergent spherical sound wave.

A point source is almost never encountered in pure form in architectural acoustics and acoustics of machines. However, in many cases, such as in evaluation of the sound radiation of machines of small dimensions, these may be considered as point sources of a corresponding output, which greatly simplifies computation. This type of computation of the acoustic field of a machine, taking into account the sound absorbing properties of a room, is shown in examples 28 through 30.

The next simple type of radiator is the piston membrane, or piston in a rigid wall (Figure 23-c). The term "piston" in the given case indicates merely that as in the previous types of radiators all the points of the radiating element vibrate in synphase and in a direction perpendicular to its surface. The sound pressure is distributed unevenly in the space surrounding this kind of radiator. At a distance r, considerably exceeding the radius a of the piston, the amplitude of the sound pressure is equal to:

$$p_a \approx \frac{f\rho}{r} \pi a^2 \dot{y}_a \Phi(\varphi, a, f). \tag{72}$$

Here φ is the angle between the axis of the piston and the direction to the point at which the pressure is determined; the other symbols, except Φ, have been introduced earlier.

The factor Φ is a certain function of the parameters φ, a, and f, indicating the degree of directionality of the radiation. The directionality of radiation is determined by the addition of the vibrations emanating from various points of the radiator and the solid wall surrounding the radiator.

For frequencies at which the radius of the piston is considerably smaller than the wave length of the radiated sound, $\Phi(\varphi, a, f) = 1$. In this case radiation is not the same in all directions, i.e., the piston is equivalent to a point source, with the only difference that at equal amplitudes of volumetric velocity the magnitude of sound pressure in the field of the piston is twice as great as in the field of a point source. Bearing in mind the fact that $\pi a^2 \dot{y}_a$ is the volumetric velocity or the output of the source, indicated in the foregoing by Q_a, the above may be readily understood. The doubling of the pressure is caused by the reflection of sound from the solid wall.

If the wall were absent and the piston, as before, radiated to one side (i.e., the piston is situated in the open end of a long tube), a doubling of pressure in the medium would not be noted, and a small piston would be completely equivalent to a point source. A condition of this kind occurs in the radiation of sound from exhaust pipes: the mass of gas in the exit section of a tube of this type may be simplified and equated to a vibrating piston.

At higher sound frequencies, where the dimensions of the piston coincide with the wave length of the sound in the surrounding medium, the picture of radiation will be substantially different. In this case the equation $\Phi(\varphi, a, r) = 1$ is fulfilled only for the angle $\varphi = 0^{\circ}$, i.e., the equation of sound pressure characteristic of a point-source radiator, valid only on the axis of the piston. The main radiation is concentrated in a cone, the angle φ_0 at the apex of which may be found with the equation:

$$\sin \varphi_0 = 0.6 \frac{\lambda}{a}, \qquad (73)$$

where λ is the length of the sound wave; at $\varphi = \varphi_0$ the sound pressure $p_a = 0$.

Within this so-called main maximum of radiation are observed concentrically-spaced additional maximums, the intensity of which is considerably lower than the main maximum. The properties of directional radiation must be studied in the course of experimental investigation of a noise which is formed as a result of the vibrations of various types of diaphragms, membranes and small rigid panels.

Although the types of radiators described in the above include a considerable number of sound radiators actually encountered, the radiation produced by the vibration in bending of plates included in the bounding structure of compartments is of greatest interest in architectural practice. This type of radiation (Figure 23-d) not only is very important but also is very complex, and because of this has been subjected to the least research. During recent years, due to the works of L. Cremer, K. Gesele, L. Ya. Gutin, et al., the problem of sound radiation of plates in which curved vibrations are excited has been clarified to a definite extent. We present a general outline of the character of this type of radiation, referring interested parties to more detailed expositiens of the problem in the appropriate sources (33, 70, 124, 132).

Let us assume that a sinusoidal curved wave is excited in a very extensive (theoretically infinite) plate with amplitude of the speed of vibration $\dot{y}_a$ (Figure 23-d). If the wave in the plate is a travelling wave, radiation into space occurs at an angle γ, for which

$$\sin\gamma = \frac{\lambda_a}{\lambda_p},$$

where λ_a and λ_p are the length of the sound wave in the medium (in this case, air) and the length of the wave of bending vibration in the plate (71), respectively.

From the theory of vibrations it is known that a standing wave in any given vibrating system may be represented as the result of addition of two moving waves extending in opposite directions. It is apparent that in the presence of a standing bending wave in the plate, radiation will take the form of two planar waves amanating at angles γ to the normals for which:

$$\sin\gamma = \pm\frac{\lambda_a}{\lambda_p}.$$

The intensity of the radiated waves is equal to:

$$J = \frac{\rho c}{2}\dot{y}_s^2\frac{1}{\cos\gamma} = \frac{\rho c}{2}\dot{y}_s^2\frac{1}{\sqrt{1-\left(\frac{\lambda_a}{\lambda_p}\right)^2}}. \qquad (74)$$

The expression for J is real only at $\lambda_a \leqslant \lambda_p$. At $\lambda_a \ll \lambda_p$, i.e., at very high frequencies,

$$J = \frac{\rho c}{2}\dot{y}_s^2,$$

and as is the case for an infinite piston membrane, the plate radiates a planar wave in a direction very near to normal.

For the purpose of examining in detail the character of radiation at various frequencies we express the frequency factor of radiation in formula (74) in decibels:

$$S = 10\log\frac{1}{\sqrt{1-\left(\frac{\lambda_a}{\lambda_p}\right)^2}}. \qquad (75)$$

In Figure 24 this relationship is depicted by the solid line. At $\lambda_a \gg \lambda_p$ the expression under the sign of the logarithm is minimal, i.e., there is no radiation. As will be seen in the following, this

condition corresponds to an abscissa value less than unity. This is explained physically by the fact that at a small wave length of the bending wave in a plate, various parts of its surface, vibrating in counter-phase, continuously "pump over" energy from one portion of the medium next to the plate to another.

At $\lambda_a = \lambda_p$ the radiation of a very extensive plate increases without limit (cf. formula 75). From formula (74) and the expression for $\sin\gamma$, it is apparent that $\cos\gamma = 0$, i.e., at any given point in semi-space the radiated wave progresses parallel to the surface of the panel. The frequency corresponding to the equality of wave lengths is called the critical frequency. On the basis of expressions for the length of longitudinal and bending waves included under Chapter 1, we obtain the expression for the critical frequency f_{kr} in the form:

$$f_{kr} = \frac{c^2}{2\pi h}\sqrt{\frac{12\rho(1-\mu^2)}{E}}; \qquad (76)$$

where h is the thickness of the plate;

ρ, E and μ are the density, modulus of elasticity and coefficient of Poisson, respectively, of the material of the panel;

c is the speed of sound in the medium surrounding the panel.

The critical frequency for a steel plate in air is approximately

$$f_{kr} \cong \frac{1250}{h}, \qquad (77)$$

where h is expressed in centimeters.

In the following chapters the values of the critical frequency are indicated for plates of other materials, in the course of discussion of the phenomenon of sound insulation, because the concept of this frequency is no less important in problems of sound radiation than in problems of sound insulation. This is due to the fact that resonance of coincidence (124) occurs at the critical frequency, and at this frequency the plate or partition transmits sound (if there is no mechanical loss in them). The resonance of coincidence is essentially a phenomenon comprising the inverse of the radiation at an angle γ of sound by a plate, mentioned above. If a planar wave impinges on the panel at an angle $\gamma = \arcsin\frac{\lambda_a}{\lambda_p}$ (in a diffuse sound field such

angles of incidence of sound always are found), a resonant wave process arises in the panel and it transmits the wave without diminishing it, also resulting in loss of sound insulation of the panel.

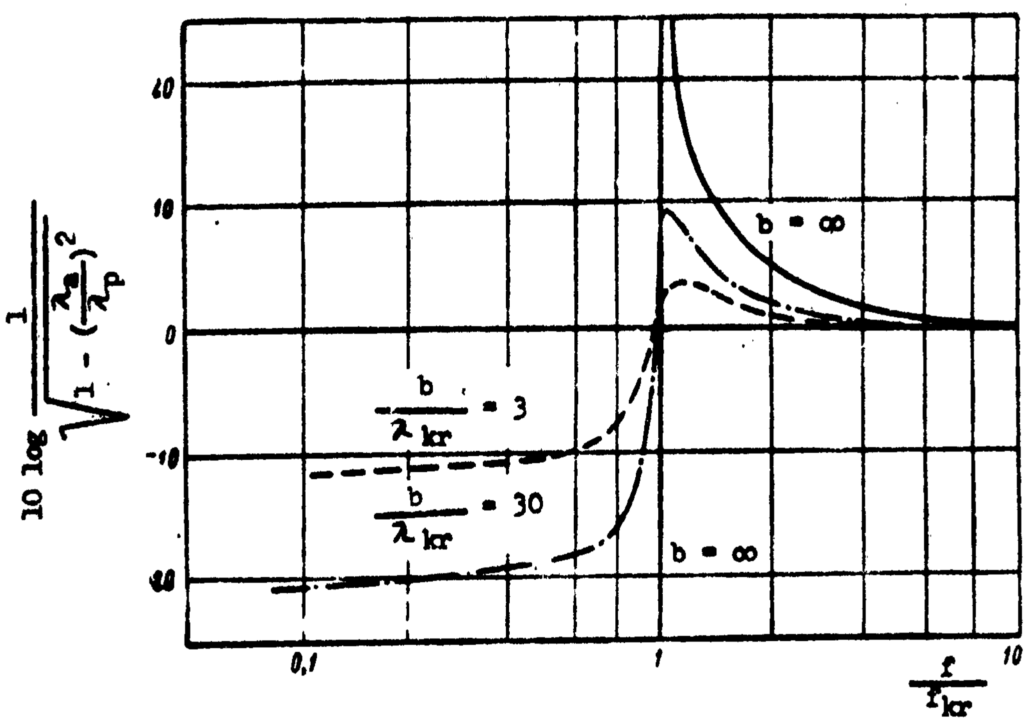


Figure 24. Frequency function of the radiation factor for bending vibrations of a plate.

———— Infinite plate;

- - - - - and -.-.-.-.-.- plate with finite dimensions.

The sound radiation functions described in the foregoing are characteristic of infinite plates undergoing bending vibration. In plates of finite dimensions marked radiation is observed also in the range below the critical frequency. Let us indicate the dimension of the plates in the direction of propagation of the bending wave by b, and indicate the wave length at critical frequency by λ_{kr}. Figure 24 indicates the values of the factor of radiation of the panel as a function of the frequency and the ratio $\frac{b}{\lambda_{kr}}$ (132). The values $\frac{b}{\lambda_{kr}} = \infty$ or $b = \infty$, i.e., an infinite plate, correspond to the solid line mentioned in the foregoing. Radiation in the subcritical range increases in proportion to a decrease of $\frac{\lambda b}{\lambda_{kr}}$. At a value $\frac{b}{\lambda_{kr}} = 3$ the difference in the level of sound radiated by the plate at high and low frequencies does not exceed 10 to 15 db.

The curves of Figure 24 relate to radiation by standing bending waves in plates of small dimensions. However, an analogous relationship holds true for travelling waves, also, although idealized pictures of the radiation (radiation through openings of finite dimensions in an infinite screen, covering a plate in which a travelling wave is being propagated) must be employed to satisfy the conditions of finiteness of the plate.

Another cause of the mutual approximation of the levels of radiated sound at high and low sound frequencies is the damping of vibrations of bending of a plate due to external or internal friction. Here also, as a result of diminution of vibrations in the plate marked radiation occurs only in a limited portion of the plate, accompanied by the appearance of radiation in the subcritical range of frequencies.

At frequencies f lying within the limits

$$f_0 < f < f_{kr} \tag{78}$$

where f_0 is any value having a frequency dimension and equal to $f_0 = \frac{c}{b}$ (c is the speed of sound), the radiation factor may be represented in the form

$$S = 10 \log \frac{1}{\pi} \; \frac{f_0}{f_{kr}} .$$

For a steel plate in air

$$S = 10 \log \frac{1}{\pi} \frac{c}{b} \frac{h}{1250}, \qquad (79)$$

where the dimensions h and b are given in centimeters, and the speed of sound in cm/sec.

From the above it is apparent that within definite limits the frequency of sound energy radiated by a plate vibrating in air is proportional to the thickness of the panel. This must be taken into account in comparative evaluation of the amount of sound radiation contributed by various structures of differing thickness, but vibrating with the identical amplitude (for example, the thick walls of engine cylinders and portions of a thin-walled engine housing).

Example 19. The thickness of a steel plate vibrating in air is 5 mm, and the transverse dimension is 50 cm. The problem is to determine the frequency at which increased radiation of sound by the plate is to be expected, and the approximate range of frequencies in which the radiated sound energy will be proportional to the thickness of the plate.

Solution. According to formula (77) the critical frequency for a steel panel in air is equal to:

$$f_{kr} = \frac{1,250}{h} = \frac{1,250}{0.5} = 2,500 \text{ cycles.}$$

Beginning with this frequency, radiation occurs with great intensity. Within the limits

$$\text{from } f_{kr} \text{ to } f_0 = \frac{c}{b} = \frac{34,000}{50} \cong 680 \text{ cycles,}$$

the energy radiated by the plate is directly proportional to the thickness of the plate.

12. Interference and Diffraction of Sound

The principle of superposition may be applied to sound waves, according to which each sound vibration occurs independently of other vibrations at a given point of space. The superposition of two or more vibrations upon each other is called interference. This phenomenon may occur for either longitudinal or transverse vibrations (in solid bodies).

During the period of sinusoidal vibration the phase of vibration at each point of space changes continually. If two vibrations occur in the same phase, strengthening of vibration is observed. In the case of movement of particles in antiphase, i.e., in a different direction, the vibration is weakened.

When the amplitude of interfering vibrations occurring in phase is identical, the total vibration is doubled in amplitude; when they are superposed in opposite phase the amplitude of the total vibration is equal to zero. This situation occurs in the formation of standing waves in any given sound conductor, such as tubes, rods, plates or columns of air.

A very peculiar phenomenon occurs when two vibrations with a certain amount of difference in their frequencies are superposed. In this case a beat occurs, markedly noticeable to the ear. The frequency of the beat is equal to the difference in frequency of the interfering vibrations. The sound, within the composition of which the beat occurs, is very unpleasant to hear, as are other sounds which vary in strength.

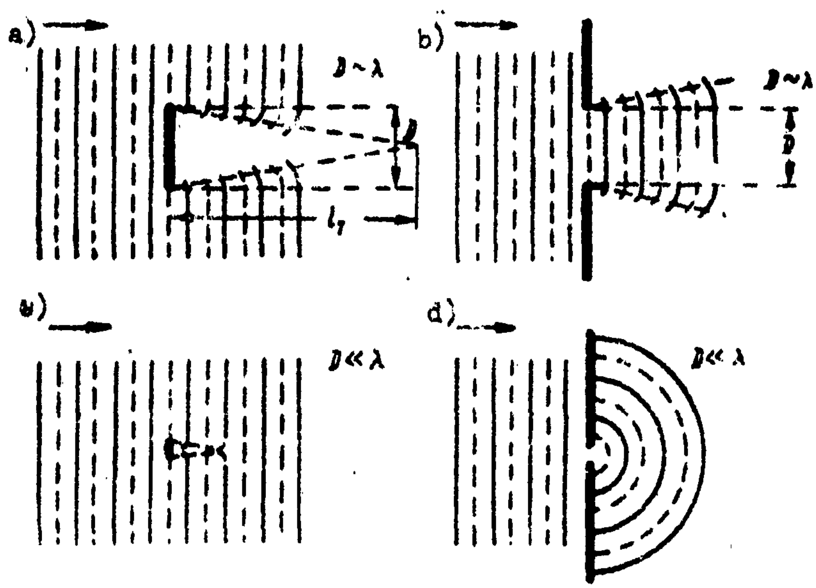


Figure 25. Diffraction of sound waves involving screens and apertures of various sizes.

At the present time interference is utilized in noise reduction technology for local reduction of sound at individual points in the compartments of transport vehicles.

The phenomenon of diffraction also is encountered frequently in practice, consisting of the bending of sound waves by obstructions. Diffraction may be easily explained in that similarly to the principle of Huygens, each vibrating particle of the medium in which sound is propagated may be considered as an elementary source of spherical waves. As a result part of the wave energy penetrates into the region which is in the shadow of the obstruction.

The degree of penetration of sound waves into the region shaded by the obstacle depends upon the ratio between the size of the obstacle and the length of the sound wave. The longer the wave length, the less is the penetration of the shadowed area with a given dimension of the obstruction (Figure 25, a and c). The length of the zone of shadow ℓ_s behind an obstacle with transverse dimension D may be computed according to the following formula:

$$\ell_s = \frac{D^2}{4\lambda} + \frac{D^2 f}{4c} . \tag{80}$$

This formula relates to diffraction in parallel rays. When sound waves originate from a single point in front of the screen the length of the zone of shadow behind it increases. Figure 26 contains a graph for computation of the decrease in intensity of sound, in decibels, at a point located at a distance D from a screen with height H. The source is at a distance R from the screen. The quantity N is plotted on the horizontal axis of the graph, and is equal to:

$$N = \frac{2}{\lambda}\left[R\left(\sqrt{1+\left(\frac{H}{R}\right)^2}-1\right)+D\left(\sqrt{1+\left(\frac{H}{D}\right)^2}-1\right)\right] \tag{81}$$

This relationship is useful in designing various sound-protection screens.

With the passage of sound through apertures or perforations, diffraction causes propagation of the sound not only along the axis of the aperture, but also laterally from the axis. The formal picture in this case is similar to the picture of radiation of a piston in a screen (Paragraph 11). Just as in the case of a piston radiator, when the dimensions of the aperture are considerably smaller than the wave length the radiation in the semi-space behind the partition is

nondirectional (Figure 25-d); if the dimensions of the aperture are identical with the wave length or greater than the latter, radiation behind the partition is localized in a more or less narrow beam (Figure 25-b).

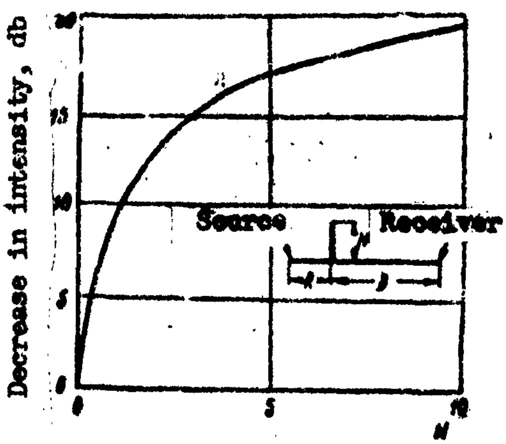


Figure 26. Weakening of sound bent by a screen, as a function of the height of the screen and its distance from the source and receiver of the sound.

Diffraction results in intense passage of sound through apertures or performations, which is the cause of reduction of the sound insulation of partitions. As a result of diffraction the sound field is distorted when an obstruction of dimensions equal to the sound wave is installed in it. This requires a diffraction correction in measuring the sound field of large microphones. The magnitude of correction may attain 3 or 5 db, or more.

13. Reflection of Sound

Reflection of sound to a considerable extent resembles reflection of light. As in the case of light, in the absence of dispersion, sound is reflected from a surface at the same angle as the angle of incidence (Figure 27-a and 27-b).

From this it follows that convex surfaces disperse sound (Figure 27-c), and concave surfaces focus and concentrate sound (Figure 27-d).

Let us assume that a sound wave impinges normally on the line of division between two media (Figure 28). The energy of the impinging sound is equal to the sum of the energy of the reflected sound and sound passing through the second medium:

$$J_{imp} = J_{ref} + J_{trans},$$

whence

$$\frac{J_{ref}}{J_{imp}} + \frac{J_{trans}}{J_{imp}} = 1.$$

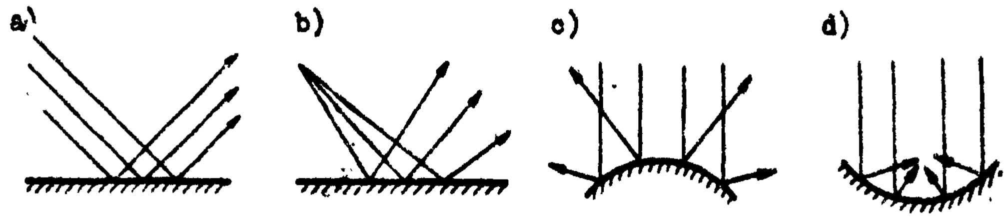


Figure 27. Reflection of sound from surfaces of various shapes.

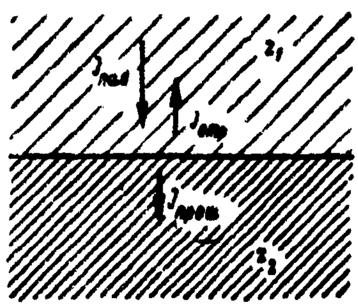


Figure 28. Reflection of sound from the surface of discontinuity between two media.

(1) J_{imp} (impinging)

(2) J_{ref} (reflected)

(3) J_{trans} (transmitted)

The first fraction may be called the coefficient of reflection of sound energy. We shall indicate it with the symbol ϑ. The second fraction is the coefficient of transmission or absorption of sound α. Thus:

$$\vartheta + \alpha = 1. \qquad (82)$$

For the coefficient of reflection of sound impinging normally upon the border of the media, we have the expression:

$$\theta = \left(\frac{z_1 - z_2}{z_1 + z_2}\right)^2 = \left(\frac{1 - \frac{z_2}{z_1}}{1 + \frac{z_2}{z_1}}\right)^2, \qquad (83)$$

where z_1 and z_2 are the specific acoustic resistances of the media.

Very frequently an expression for the coefficient of reflection of sound with reference to pressure or speed of vibration is used. It is equal to:

$$\vartheta' = \sqrt{\frac{J_{ref}}{J_{imp}}} = \left|\frac{z_1 - z_2}{z_1 + z_2}\right|. \qquad (84)$$

Taking into account the relationships (82) and (83), we may find the coefficient of transmission of sound through the boundary between the two media. The coefficient of transmission of sound energy is equal to:

$$\alpha = 1 - \left|\frac{z_1 - z_2}{z_1 + z_2}\right|^2 = \frac{4\,|z_1 z_2|}{|z_1 + z_2|^2}. \qquad (85)$$

The coefficient of transmission of sound pressure is found by taking into account the relationship between the sound energy and sound pressure in each of the media. We thus obtain:

$$\alpha' = \sqrt{\alpha\frac{z_2}{z_1}} = \frac{|2z_2|}{|z_1 + z_2|} \qquad (86)$$

Figure 29 is a graph of the relationship between the coefficients of reflection and transmission of sound as a function of the ratio of the acoustic resistances of the media (formulas 83 and 85). The value of the ratio $\frac{z_2}{z_1}$ is located on the horizontal axis, in a logarithmic scale. From the graph and from the formula it may be seen that the value of the coefficient of reflection of sound from the boundary between two media depends only upon the absolute value of the ratio of the acoustic resistances of the media but does not depend upon which of the resistance is greater. Because of this the sound propagating through any given solid thick wall undergoes reflection from the boundary of the wall and air identical to the reflection exhibited by sound propagating through air and reflected by that wall.

Let us examine an example of the utilisation of this relationship in the acoustics of ships. A liquid (water, oil) conducts sound well, and has great acoustic resistance in comparison to air (Table 1). Because of this, if fuel or water tanks of a ship, within which sound vibrations from machinery installations are propagated, are separated from housing areas of the ship by an air layer (cofferdam) in a manner ensuring that vibrations will be reflected from this layer, the acoustic regime of the occupied areas will be improved. An air layer between two rigid walls having considerable acoustic resistance has analogous significance in the insulation of air sounds.

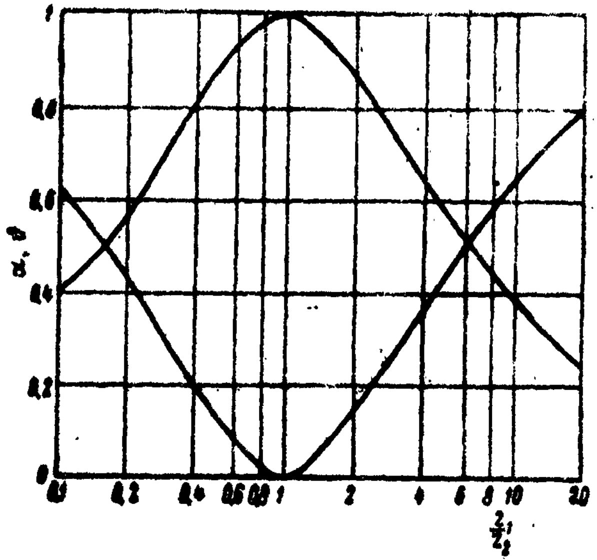


Figure 29. Coefficient of reflection (lower curve) and transmission of sound (upper curve) at the boundary between two media as a function of the ratio of the acoustic resistances of the media.

Formulas (83) and (84) were discussed primarily in relation to the reflection and penetration of sound at the boundary between two homogenous, infinite media. However, these formulas also apply with certain limitations to any similarly dimensioned sound conductors if z_1 and z_2 are considered as the acoustic resistances of semi-infinite portions of this sound conductor. Let us consider in particular the application of formula (84) in determining the reflection of sound under the conditions of change in the cross-section of the sound conductor (Figure 30).

The acoustic resistance of individual portions of a sound conductor (formula 50) is: $z_1 = \rho c S_1$; $z_2 = \rho c S_2$.

Here S_1 and S_2 are the cross-sectional areas of each portion.

On the basis of equation (84) the coefficient of reflection is:

$$\vartheta' = \left| \frac{1 - \frac{S_2}{S_1}}{1 + \frac{S_2}{S_1}} \right|. \qquad (87)$$

This value of reflection holds for both the passage of sound from a wider section into a narrower one (Figure 30-a), and vice versa (Figure 30-b), if the difference in the values S_1 and S_2 is not exceptionally great. Another condition of the application of formula (87) is a small cross-sectional area of the sound conductor in comparison to the wave length passing through it.

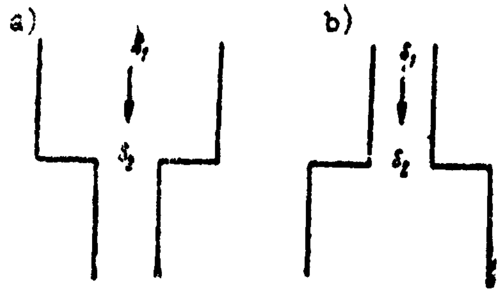


Figure 30. Reflection of sound under the condition of change in the cross-section of the sound conductor.

Example 20. The problem is to determine the coefficient of transmission of sound pressure with normal impingement of a sound wave from air upon the steel wall of a tank filled with water (sound frequency 500 cycles, and wall thickness 4 mm).

Solution. We apply formula (86), taking account of the fact that in this case

$$z_1 = \rho c_{air};$$

$$z_2 = j\omega m + \rho c_{water},$$

where ωm is the inertial resistance of the wall at the given frequency; the subscripts $_{air}$ and $_{water}$ in the expressions for wave resistance refer to the medium.

The impedance of the wall at a frequency of 500 cycles is:

$$\omega m = 2\pi \cdot 500 \cdot 0.4 \cdot 8 = 10^4 \gg \rho c_{air}.$$

Thus the value of z_1 in the denominator of (86) may be ignored in comparison with z_2. Thus we obtain:

$$\alpha' \cong \frac{2\,|j\omega m + \rho c_{water}|}{|j\omega m + \rho c_{water}|} = 2,$$

i.e., the pressure is doubled upon passing through the wall. However, as may be deduced from formula (85), the energy passing into water is negligible, and the large value of the transmitted sound pressure results only from the large acoustic resistance of water. It may be stated, also that when a diffuse sound field obtains in front of a thin wall, the pressure in a liquid on the other side of the wall is not double the value of the pressure, as in the case of a free sound wave impinging on a wall, but is equal to the value of pressure in the diffuse field.

The same steel wall, not bordering on water but situated in air, ensures a considerable weakening of sound pressure (as a result of very low resistance of the medium behind it); cf also formulas (100), (101), etc.

CHAPTER 4

THE MEASUREMENT OF NOISE AND SONIC VIBRATION

14. Instruments for Measuring the Level of Airborne Noise

Noise meters are divided into subjective and objective types according to the operating principle and construction of these instruments for measuring the level of sound and noise.

In subjective noise meters, or phonometers, the measured sound or noise is compared with a pure tone of definite frequency, produced by a special generator. In the course of measurement the operator applies a radiator of a sound, connected by means of an acoustic circuit to the tonal generator (Figure 31), to one ear, and receives the sound to be measured in the other ear. With the aid of the strength regulator (attenuator) of the noise meter, equality of perception of the loudness of the sound from the generator and the measured sound may be obtained.

Because the hearing apparatus of man participates in the measurements, the levels thus obtained are the true levels of loudness of the sound. The values of these levels are computed on the scale of the attenuator, calibrated in phons.

The possibility of obtaining values directly characterizing the degree of the effect of a sound upon the human ear is the advantage of the phonometric method and instrument. The disadvantages include the complexity of the process of measurement, and dependence of the results upon the hearing characteristics of each operator. Because of this phonometry is not employed in ordinary acoustic measurements, but only in certain research work.

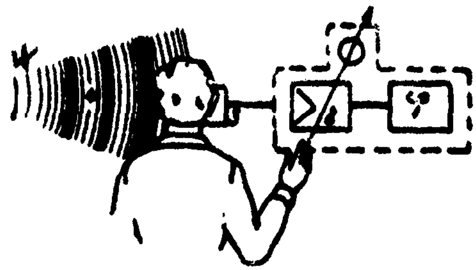


Figure 31. Diagram of measurement of the level of loudness of noise with the aid of a phonometer.

1 - Sound generator
2 - Amplifier, with logarithmic attenuator
3 - Earphone
4 - Measured sound

Objective noise meters have found considerably more extensive application. In these instruments (Figure 32) the measured sound is perceived with the aid of a microphone, which has a broad range of working frequencies. Electrical vibrations from the microphone are amplified and transmitted to a circuit consisting of capacitors, resistances and inductances, and forming the frequency characteristics of sensitivity of the instrument into frequency curves of equal loudness.

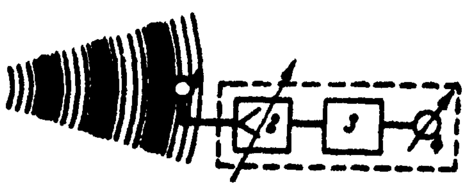


Figure 32. Block diagram of an objective noise meter.

1 - Microphone, or pick-up
2 - Amplifier with attenuator
3 - Frequency correction link
4 - Measuring instrument

However, the most highly perfected noise meter cannot produce curves of equal loudness for any given value of intensity of sound. Most often noise meters are limited to three scales, corresponding to curves of equal loudness for medium and high values of sound intensity. Types of noise meters with a large number of scales also have been proposed.

Because of its limited number of frequency characteristics of sensitivity in comparison with the human ear, the objective noise meter enables determination of only approximative values of the loudness level of sound. As mentioned in the foregoing (cf. Table 4) these approximative values of the level of loudness of sound

are called sound levels, and are expressed in decibels. In measuring complex sounds the value of sound level registered by the noise meter may differ by 6 to 8 db from the true value of the level of loudness of a sound (usually below the level).

Figure 33 shows two models of portable precise noise meters incorporating semiconductors, and Figure 34 depicts the frequency characteristics of sensitivity of the precise noise meter of the firm Bruel and Kjaer (Denmark). This noise meter has four frequency characteristics of sensitivity, indicated by A, B, C and Linear. The first characteristic is used at noise levels up to 60 db, the second within the range of 60 to 130 db, and the third at levels above 130 db. The linear characteristic is utilized in measuring sound pressure.

a)

b)

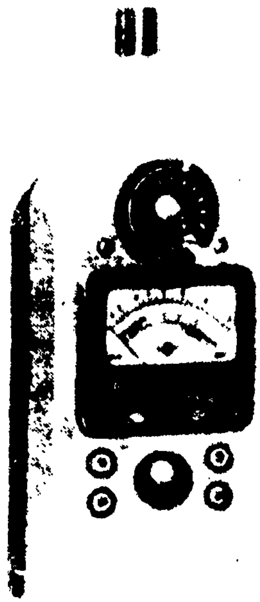


Figure 33. Portable objective noise meters incorporating semiconductors.

Portable objective noise meters also have been developed by Yu. M. Il'yashuk and I. I. Slavin (96) at the Leningrad Institute of Labor Safety.

The microphone of the noise meter occasionally is provided with an extension cord, enabling measurement of noise at various points of a room without moving the instrument from place to place.

Figure 35 illustrates the measurement of noise of a machine with the aid of an objective noise meter. The noise level is determined by summation of the indications of the logarithmic amplifier switch and of the needle indicator of the instrument.

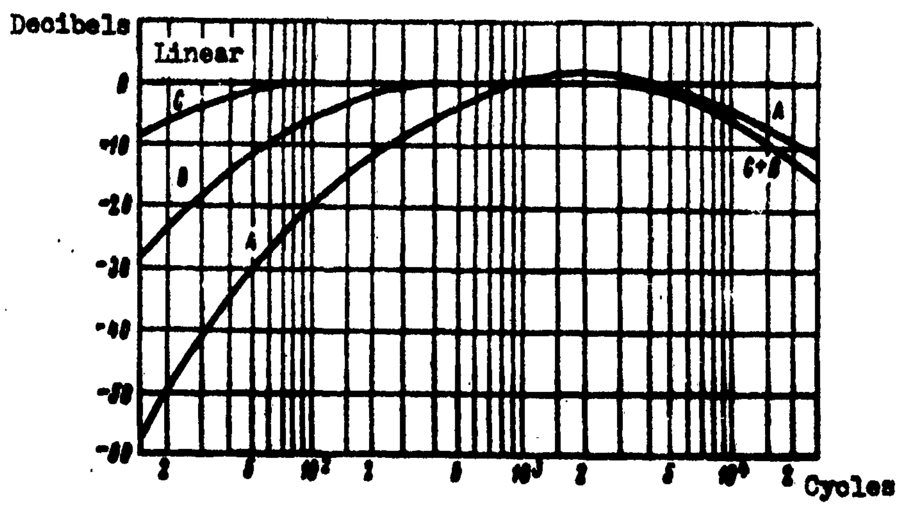


Figure 34. Frequency characteristics of a noise meter.

It must be mentioned that measurement of the level of unpleasantness, or degree of irritating effect of noise may be no less important than measurement of the level of loudness of noise. For this purpose the frequency characteristics of sensitivity of the instrument measuring noise should correspond not to curves of equal loudness, but to curves of equal unpleasantness of sound (Figure 17). Instruments of this type have not been developed as yet, and the only instrument available, characterizing the effect of noise on the human organism is the noise meter. It may be noted that the use of a noise meter with one-third-octave or one-half-octave filters enables determination of the role of high-frequency components in the spectrum of an investigated noise, together with approximate evaluation of its unpleasantness.

Figure 35. Measuring the noise of a machine with the aid of an objective noise meter equipped with microphone on an extension cord.

35. Instruments for Measurement of Sonic Vibrations

Measurement of the parameters of the process of sonic vibration, or, what is the same thing, the level of vibration, is required in many instances of shipbuilding and architectural practice: e.g., in investigation of the phenomena of propagation of vibrations in the hull and in foundation structures; in locating resonant portions and par of these structures radiating strong noise; and in evaluating the effect of any given measures taken for controlling noise.

A special instrument, the vibrometer, is used in measuring sonic vibration. The term "vibrometer" has two applications. This term is taken to mean the sensing instrument detecting a vibration, i.e., vibration detector. Most often the term vibrometer is taken to refer to the entire apparatus for measurement of vibration including, in addition to the detector, the amplifier and registering portions of the instrument (in this case the vibration sensor itself may be called a vibration pick-up, in conformity with terminology frequently applied in practice). However, a tendency was noted toward the development of instruments intended solely for measurement of vibration (for example, the "General Radio" vibrometer in the US), which at the present time has given way to

a trend in production of instruments serving simultaneously for the measurement (and analaysis) of noise and vibration. An over-all view of an instrument of this type (27) is shown in Figure 36. A symplified block diagram of a circuit for measurement of sound vibration is presented in Figure 37.

All modern designs of vibration pick-ups in the sonic range are based on the electroacoustic principle, i.e., on the principle of transformation of the mechanical energy of the vibrations which are being measured into electrical energy. The application of this principle is a result of the broad potentialities of electrical designs for amplification and analysis of very weak vibrations which, however, and as we have seen in the foregoing, may evoke very unpleasant physiological effects.

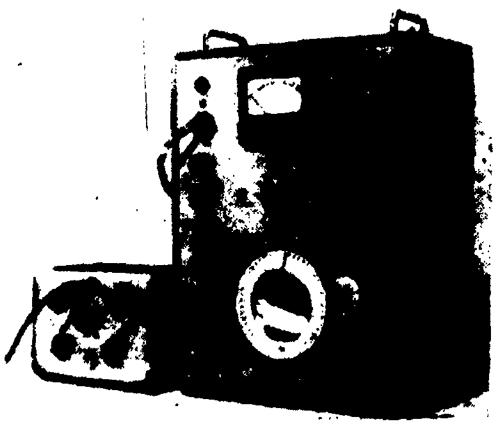


Figure 36. Vibration (and noise) analyser.

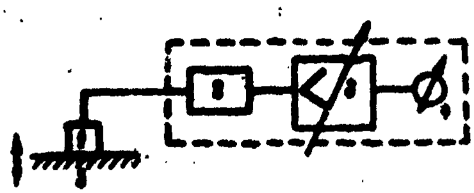


Figure 37. Block diagram of a vibrometer.

(1) Vibration pick-up
(2) Integrator
(3) Amplifier, with attenuator
(4) Measuring instrument
(5) Vibrating surface

Conversion of mechanical vibrations into electrical vibrations occurs in the vibration pick-up, itself. The vibrations are amplified in the electrical link of the vibrometer and are forwarded to the indicating apparatus, which most often consists of a needle indicator. The amplifier of the vibrometer, like the amplifier of a noise meter, is provided with an amplification regulator, calibrated in logarithmic units. To the electrical portion of the vibrometer may be attached an earphone for listening to the vibrations, and instruments for recording and for frequency analysis of the vibrations.

Many systems for conversion of the mechanical vibr ions into electrical ones exist, and correspondingly numerous syste s of vibration pick-ups have been developed. In shipboard vibrometric practice, however, vibration detectors of the piezoelectric type, which are suitable and adequately efficient for this type application, are used almost exclusively. Their action is based on the piezoelectric effect, consisting of the generation of electrical charges on the faces of crystals and wafers of certain materials when static or intermittent forces are applied to them. The materials of this type (piezoelectric materials) include Rochelle salt, barium titanate, ammonium phosphate, quartz, tourmaline, etc.

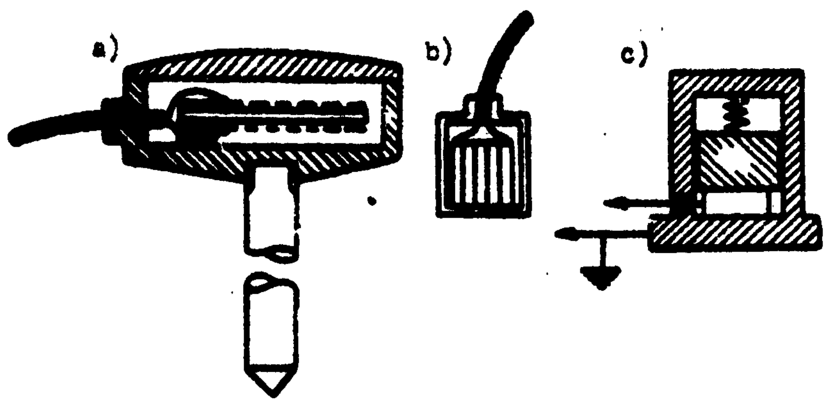


Figure 38. Construction of piezoelectric vibration pick-ups.

Diagrams of the structure of three piezoelectric vibration pick-ups are shown in Figure 38. In the vibration detector of design shown in Figure 38-a, which is produced by many foreign firms and still is used fairly extensively in the Soviet Union, the sensitive element is a wafer, or a combination of a wafer of Rochelle salt of relatively large size, fixed by means of a bracket within the housing of the vibration detector. The vibration of a surface upon which the pick-up is placed is transmitted through a vertical rod to the housing and the location of the piezoelectric element, causing flexural vibration in the latter. The inertial forces arising as a result of this lead to the formation of piezoelectric charges of varying frequency at the faces. The electric voltage from the faces of the piezoelectric element are fed into the amplifier of the vibrometer. The piezoelectric vibration detector may be plugged into not only the special tube-type vibrometric circuit, but also into any noise meter with high resistance input, which uses a piezoelectric microphone.

Because the magnitude of the voltage at the faces of the piezoelement is determined by inertial force, and the latter is proportional to the vibratory acceleration at the point of application of the vibration detector, the vibration detectors of the piezoelectric system are pick-ups of the vibratory acceleration. This is why instruments using single or double integration of the indicated piezoelectric vibration sensor (Figure 37) must be applied to obtain the value of the speed of vibration of vibratory displacement of the vibrating surface.

The vibration detector (Figure 38-a) has two substantial disadvantages:

(1) a relatively high weight (up to 400 or 500 grams), as a result of which the picture of the vibration of the structure being tested may be distorted (cf. Chapter 20);

(2) narrow range of working frequencies. This range extends from a few cycles to 1 or 1.5 kilocycles.

At higher frequencies, resonance of the rod, of the housing and of the piezoelement occurs, distorting the indications of the vibrometer. At frequencies above 2 kc, i.e., in the range beyond resonance of the piezoelectric crystal, the sensitivity of the vibration detector drops sharply.

These disadvantages are absent in the case of another design of piezoelectric vibration detector, the "point" vibration detector. It is distinguished from the above by the very small size of its piezoelement, as a result of which its resonance frequency attains 15 to 20 kc and higher. The dimensions of the housing are only slightly larger than the supporting bracket of the piezoelement, and because of this all imposed distorting resonance of the housing is absent. Because of its low weight the vibrometer has practically no influence upon the vibratory regime of the structure which is being tested.

A vibration detector of similar design was used by the present author in testing noise-isolating mounts (47). The sensing element consisted of a package of Rochelle salt crystals having a natural frequency of approximately 25 cycles, working in compression (Figure 38-b).

During recent years very highly perfected "point" vibration detectors with barium titanate ($BaTiO_3$) and zirconium titanate have been developed in Denmark and in the USSR. In one of the modifications of these pick-ups, the piezoelement is loaded with a small weight, pressed against the piezoelement by a spring (Figure 38-c). Figure 39

shows a vibration detector of similar type developed by B. N. Masharskiy and L. S. Sheyb. The working range of frequencies of this vibration detector is from several cycles to 10 or 12 kc. Its weight is 20 to 30 grams, and its sensitivity is in the range of 20 mv per g (g is the acceleration due to gravity). Conclusions may be drawn with respect to dimensions of the vibration detector by comparing it with the size of a 12.5-centimeter logarithmic slide rule, shown in Figure 39. Some types of vibration detectors also are used which incorporate two identical piezoelectric wafers or ferroelectric elements, separated by a metallic electrode.

Figure 39. A "point" piezoelectric vibration pick-up.

Irregularity of the frequency characteristics of sensitivity of a point piezodetector does not exceed ± 2 db within the working range of frequencies. The curve of the amplitude of the vibratory acceleration determined by the vibration detector (with constant vibratory displacement) is parallel to the theoretical curve corresponding to equation (20) up to the very high frequency range.

Another valuable characteristic of the piezodetector is the fact that its sensitivity in the transverse direction is 10 to 15 db lower than in the direction of its axis (curve No. 3 in Figure 40). This ensures greater detection of vibrations perpendicular to the surface to which the detector is applied. Rotational vibration and vibration in a direction parallel to the plane of application of the detector are detected very weakly (if no corresponding devices are used for reception of vibrations in this direction).

It must be borne in mind that the possibilities of point vibration detectors in relation to measurement of vibration levels in a very broad range of frequencies may be realized only with the use of special means of applying the detectors (cf. Chapter 20).

Vibration detectors based on the electrodynamic (induction) principle (14, 44) are used considerably less frequently for measurement of vibration in the sonic range. These vibration detectors consist of a permanent magnet, in the gap of which a coil is suspended on an

elastic support. The magnet is placed on the vibrating surface. With the motion of the magnet the coil intersects the lines of force in the gap, and electromagnetic force is generated in it. The magnitude of this force, according to the law of electromagnetic induction, is proportional to the number of lines of force intersected by the turns of the coil per unit time. For this reason the induction vibrodetector is a sensor of vibrational velocity.

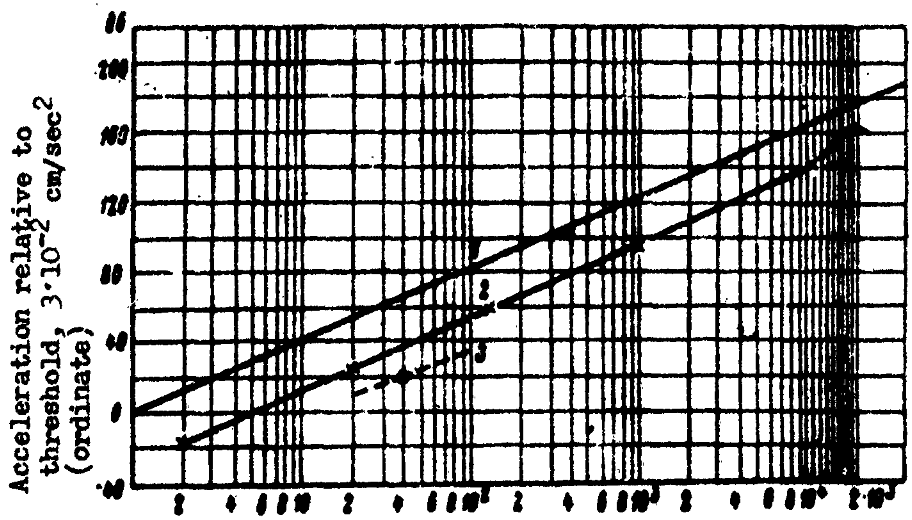


Figure 40. Frequency characteristics of the sensitivity of a "point" vibration pick-up.

(1) Theoretical curve; (2) Experimental curve taken from vibrating plate, for vibrations in the direction of the axis of the vibration detector; (3) Experimental curve of vibrations in transverse direction.

In distinction from the piezoelectric measuring sensors, the sensitive element of induction detectors has a low natural frequency, usually on the order of several cycles. The upper limit of frequency in measurements with the most highly perfected induction vibrodetectors does not exceed 5 or 6 kc. Because the magnet of the electrodynamic vibration detector has a fairly large mass, instruments of this type may be used without noticeable distortion only in the measurement of vibrations of relatively heavy parts of machines and structures.

For measurement of vibrations of low frequency, and simultaneously the amplitude of vibratory displacement, vibrographs are used. Previously, purely mechanical instruments were used almost exclusively for this purpose, i.e., instruments in which the investigated vibrations were amplified by means of a system of levers and recorded on a paper or wax tape. Vibrographs of this type were developed by Geiger, Shenk and others. With the aid of supplementary devices the mechanical vibrograph may be adapted for measurement of torsional vibration of propeller shafting. At the present time mechanical vibrographs have been displaced by instruments employing the induction system.

In the functioning of unbalanced mechanisms, especially those provided with vibration isolators, the amplitude of low frequency vibration may attain 0.5 mm or more. For determining the magnitude of such a vibration in the absence of special instruments, simple methods may be used. Figure 41 depicts a measuring wedge, which is glued to the vibrating structure in the plane parallel to the direction of vibration. Under vibration the wedge bifurcates, and the amplitude of vibration is found by computation on the basis of the wedge (marking is based on the apparent relationship of the similar triangles of the stationary and displaced wedge). With the use of a wedge with dimensions 100 x 10 mm the amplitude of vibration may be determined approximately within 0.1 to 2 mm.

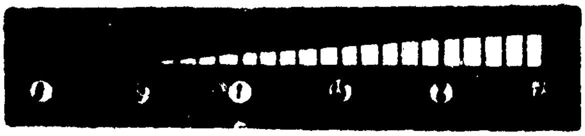


Figure 41. Measuring wedge.

16. Instruments for Recording Noise and Sonic Vibration

In acoustical practice it often becomes necessary to record a noise or vibration for the purpose of subsequent frequency analysis. Automatic registration of the level of any given parameter of investigated vibrations is of considerable interest (sound pressure, speed of vibration, vibratory acceleration, etc.). The value of this type of automatic recording lies not only in establishing an objective document of the vibratory process, but also in eliminating subjective errors of the operator. This type of recording also facilitates evaluation and comparison of the magnitudes and character of parameters of the investigated processes. Levels may be recorded automatically as functions of time, coordinates, frequency, speed of motion of the source, etc.

Recording of noise and vibrations with the aid of magnetic sound recording apparatus, the tape recorder, has very great usefulness. In this type of recording, electric vibrations from corresponding electro-acoustic transducers (microphones or vibration pick-ups) are converted into oscillations of the strength of a magnetic field, which are fixed on a special ferromagnetic tape. With reversal of the process the original sonic vibrations may be reproduced any desired number of times. The carrier tape does not change its electroacoustic and mechanical properties for a period of several years.

An outline of the principle of the tape recorder is shown in Figure 42. The tape recorder consists of two circuits, the recording circuit and the reproduction circuit. The recording circuit (left portion of the diagram) consists of a suitable electroacoustic transducer 1, an amplifier 2, and recording and erasing heads 3 and 4. The amplified oscillations are fed to the recording head, consisting of a magnet, through the gap of which passes the ferromagnetic tape-carrier 10. The density of magnetization is determined by the magnitude of the current in the coil of the magnet.

Immediately in front of the recording head is located the erasing head, which may work simultaneously with the recording head, thus enabling the use of tapes containing old recordings. Erasure of a recording is accomplished by cyclic remagnetization of the tape at a frequency of several tens of kilocycles, produced by the special generator 5. The intensity of this high-frequency remagnetization is distributed evenly across the width of the head, and upon emerging from the head the tape is completely demagnetized.

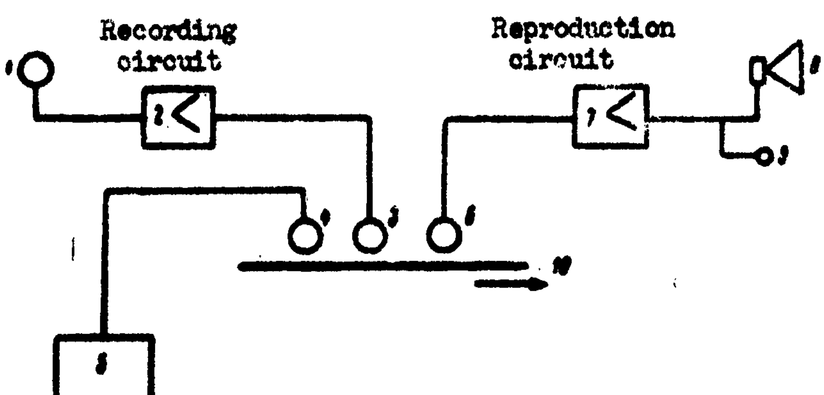


Figure 42. Block diagram of a tape recorder.

The right-hand portion of the diagram consists of the reproduction circuit. It consists of the reproducing magnetic head 6, the amplifier 7, and loudspeaker 8. At the outlet of the reproduction circuit is a socket 9, for plugging in noise analysis apparatus, earphones or automatic recorder of noise level. The tape recorder also is provided with devices for controlling the level of recording and reproduction.

a) b)

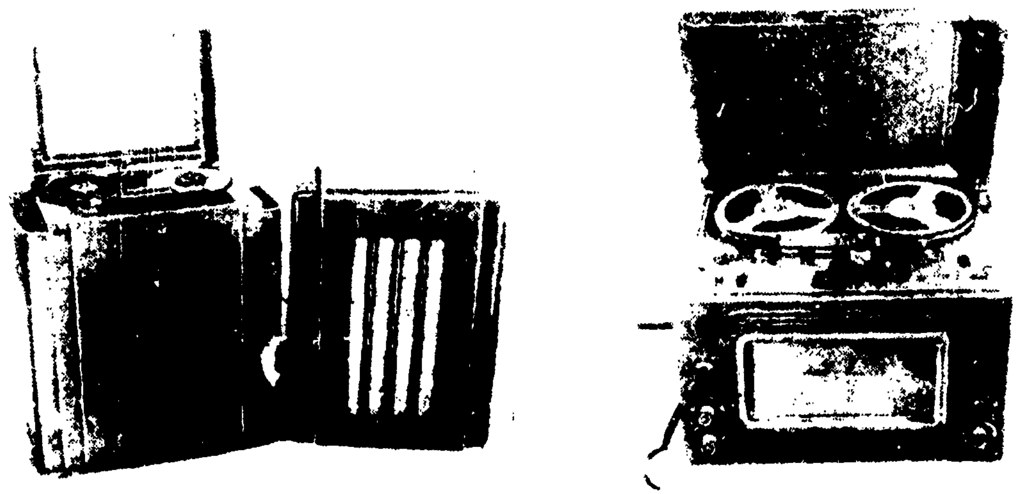


Figure 43. Tape recorders: (a) High quality stationary tape recorder; (b) Portable tape recorder.

Figure 43-a shows one of the Russian high-quality tape recorders which may be used in acoustic experiments in plants and in research organizations (its use on ships is hindered by its relatively large dimensions and weight). The tape recorder consists of two identical units for recording and reproducing (one of these units is depicted in Figure 43, plus the block of control dynamics). The working heads, control instruments and tape reels are located in the upper portion of the unit. The playing time of each reel is approximately 20 minutes, although the presence of two identical recording units enables uninterrupted recording of practically unlimited time. The tape recorder is equipped with a microphone on an extension, although the recording circuit may be connected directly to the amplifier of a noise meter or vibrometer.

In recording with a microphone the field of frequencies of the tape recorder is 50 to 8,000 cycles. In this range the irregularity of the frequency characteristic of sensitivity of the entire circuit is ± 3 db, but the dynamic range, i.e., the ratio of amplitudes of non-distorted recording, is 40 db. In portable tape recorders (Figure 43-b), which may be used on board ships, the field of working frequencies as a rule does not extend above 5 or 6 kc, and the dynamic range does not exceed 30 db.

Tape recorders for recording vibrations of infra-sonic and ultra-sonic frequencies have been developed on the basis of the types of instruments described in the foregoing.

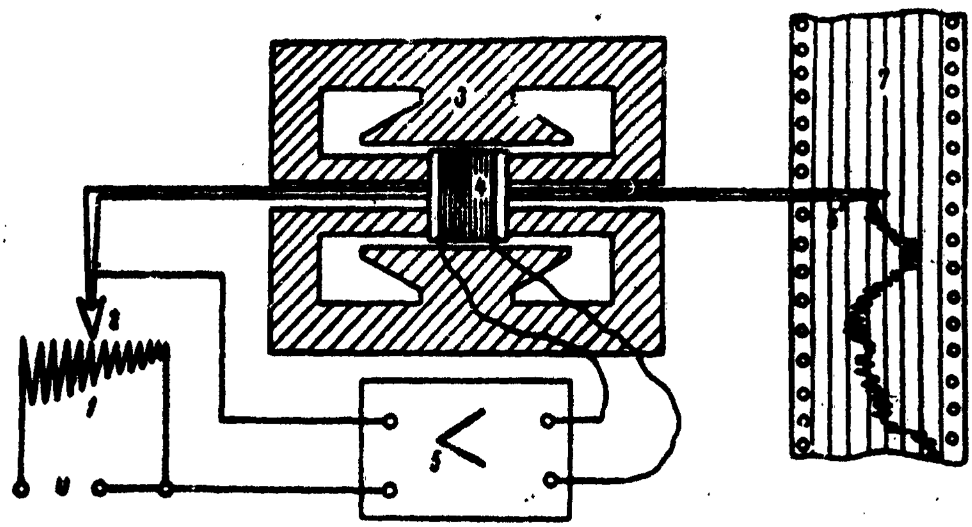


Figure 44. Arrangement of a quick-acting recorder of vibration levels.

Quick-acting recorders of level are used very extensively. They enable noise and vibration levels to be obtained in either logarithmic or linear units.

The principle of design and function of the quick-acting level recorder may be seen in Figure 44. Recorded electrical vibrations U from a microphone or vibration pick-up enter the input of the amplifier, to which a special potentiometer 1 is connected. The slide 2 of the potentiometer is connected mechanically with coil 4, located in the gap of a strong permanent magnet 3, and with the stylus of the recorder 6. The coil is connected in series with the output of amplifier 5, giving a rectified current.

Such a system of electromechanical feed-back works in the following manner. Let us assume the voltage at the input increases. The current in the final link of the amplifier and in the moving coil increases correspondingly. This results in displacement of the coil in the magnetic gap, i.e., displacement of the slide of the potentiometer to a position where a new equilibrium is established. The stylus connected with the slide depicts the increase in level on the moving tape 7. If the potentiometer is logarithmic, the recording on the tape is indicated directly in decibels.

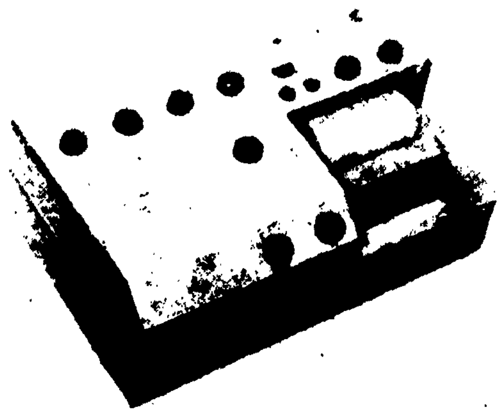
The better, contemporary quick-acting recorders (Figure 45) enable registration of changing levels of vibration with frequencies up to 20 cycles, in which the carrier frequency of the registered vibrations may comprise tens and hundreds of kilocycles. The speed of motion of the stylus of the recorder varies from 2 or 4, to 1,000 or 2,000 mm/sec, and the speed of motion of the paper ranges from 0.003 to 100 mm/sec and higher, which enables recording both slow and very fast changes in the level of the investigated process.

The possibilities of the logarithmic recorder are not exhausted by recording the level of vibratory processes. The recorder enables recordings to be made of the frequency curves of isolation and absorption of noise (cf. Chapter 8), and may be utilized as a regulator of noise level, which is very useful in many acoustic measurements. For synchronization of investigated processes according to time, and inscribing any periodic notations, the recorder is provided with a synchronized motor with contact cylinder. There is also a flexible shaft for driving external apparatus connected with the measuring system.

In addition to the magnetic recording of sound and quick-acting level recorders, train- or cathode ray tube oscillographs may be used in making acoustic measurements on ships and in industrial plants. These instruments are described adequately in the general literature on electric and acoustic measurements.

17. The Analysis of Noise and Vibration

Although the total levels of noise and vibration give fairly valuable information on the vibratory process in a room or in a structure they do not enable determination of the most intense frequency component of the process (which is necessary for evaluation of the fatiguing or masking effect of noise).



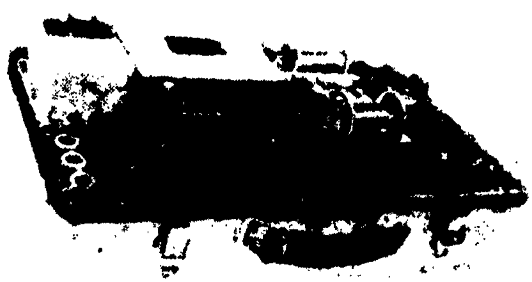


Figure 55. Quick-acting logarithmic level recorder

(a) Recorder of the firm "Bruel" and Kjaer"

(b) Russian recorder, model N-110.

In evaluating the effect of noise deadening devices, the use of only single measurements of the total level of sound or vibration may lead to substantial errors, which may be seen from Figure 56 in the following. It shows frequency curves of noise of a certain machine, taken before and after installation of a sound-absorbing structure on the machine. In the given case the noise level decreases as frequency increases, and because of this the indication of the noise meter will be determined by the levels at the lower frequencies. The effect of the application of the noise-absorbing structure at these frequencies is insignificant, and consequently the difference in the total noise levels also will be insignificant in the absence or presence of the absorbing structure. It is apparent, however, that the absorbing structure sharply reduced the level at high frequencies, thereby lowering the irritating effect of the noise.

Two types of electro-acoustic analytic devices are used for frequency analysis of noise and vibration, the filter and heterodyne types. The filter design consists of an assembly of filters; each of them eliminating a definite range of frequencies from the investigated noise. The upper and lower boundaries of the admission band, or it may be said, the range of transparency of each given filter, coincides with the corresponding boundaries of the range of admission of the neighboring filters.

The frequency characteristics of the filters are better the more they approximate a rectangular form. Filters in which the fall-off of the frequency characteristics of sensitivity over the limits of the admission band exceeds 25 db per octave are considered adequate.

The simplest analytic filtering device consists of an assembly of octave, semioctave or third-octave filters with manual switching of filters (Figure 46). As is apparent from the name of these filters, the limiting frequencies of the ranges of admission are one octave, one-half-, and one-third octave, respectively. The vibration levels in each of the bands of admission of the filters may be observed on the instrument or may be indicated on the tape of the self-recorder.

Manual switching of the filters and visual recording of levels is accompanied by drawbacks. In addition, the manual method of analysis requires a fairly large amount of time, and because of this is inapplicable in analysis of noises varying with time. This is why automatic filtering devices are acquiring ever-increasing use. Figure 47 shows the block diagram of an automatic, quick-acting filter analyzer, the spectrometer. Switching is automatic in this device, accomplished with the aid of a small motor, turning at constant speed. The speed of switching of the filter is about 15-20 times per second. In this way the apparatus records an almost instantaneous spectral picture of the noise.

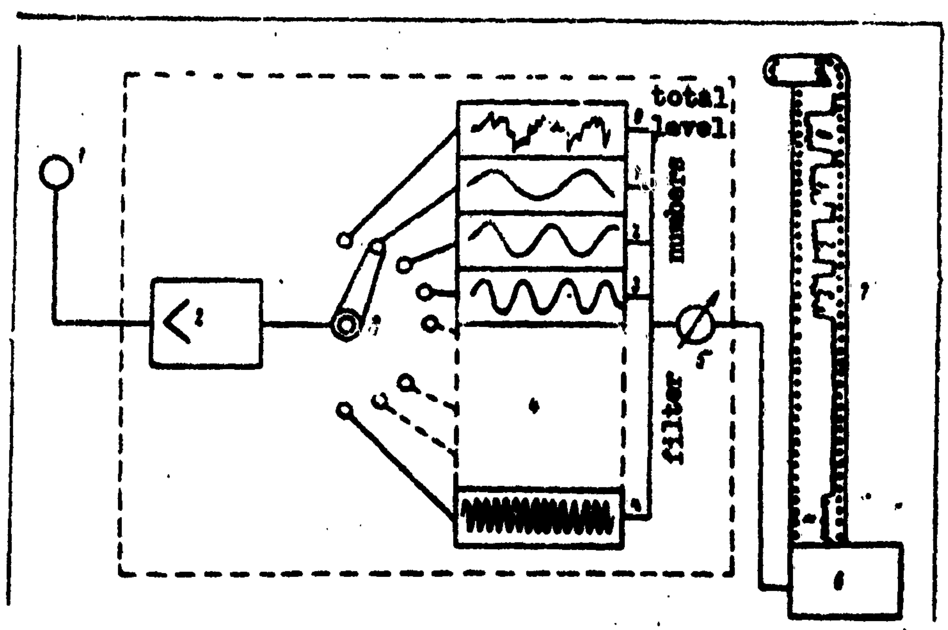


Figure 46. Block diagram of analytic filter device with manual switching of filters.

(1) Electroacoustic transducer (pick-up); (2) preliminary amplifier; (3) manual filter switch; (4) assembly of filters; (5) indicator; (6) self-recorder; (7) tape with a recording of levels within bands of admission of individual filters.

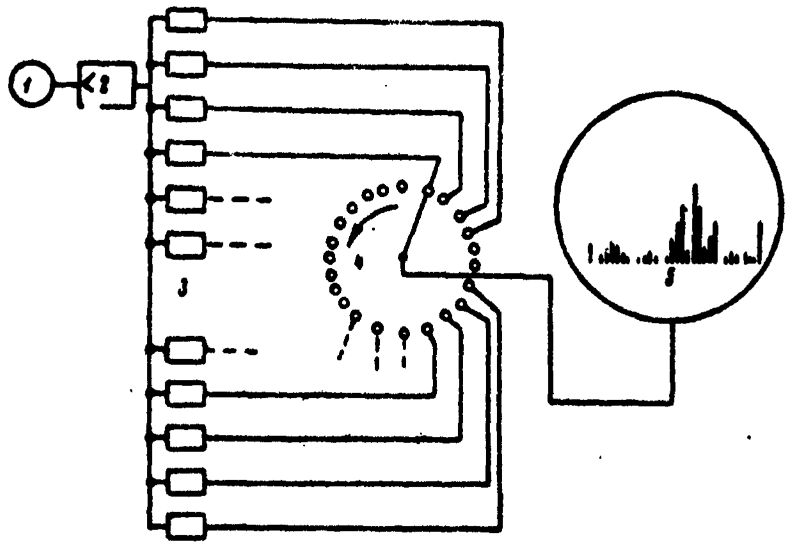


Figure 47. Block diagram of automatic filter spectrometer.

(1) Electroacoustic transducer (pick-up); (2) amplifier; (3) assembly of filters; (4) automatic switch synchronized with the scale of the cathode ray tube; (5) cathode ray tube.

The analyzed process is depicted on the scrren of the cathode tube of the spectrometer in the form of many bars, distributed on a frequency scale, the height of which indicate within a definite scale the noise level in each given band of frequency (Figure 48). The spectrum picture usually is registered with the aid of a motion picture camera, mounted against the screen of the spectrometer.

The width of the band of transparency of the filters of the spectrometer is equal to one-third, or one-half octave. When it is necessary to clarify discrete components in the spectrum of the investigated noise or, as is often said, to determine the microstructure of the spectrum, it is necessary to use an analytic instrument with greater power of resolution. Of assistance here is the principle of heterodyne transformation of vibrations, known from the field of radio electronics. Its e[illegible]nce in the given case consists mainly of feeding the output of a special generator (heterodyne) into the system containing the varying voltage produced by the electroacoustic pick-up (Figure 49). The beat of frequency difference arising from superposition

of the oscillations of the heterodyne with the investigated vibration is filtered by a narrow-band filter (such as a quartz crystal), and after amplification is fed into the registering instrument. The frequency of the heterodyne changes continuously and evenly, and thus the tone of the heterodyne ("seeking" or "sounding") passes through the entire investigated spectrum of the noise, separating the frequency components contained in it.

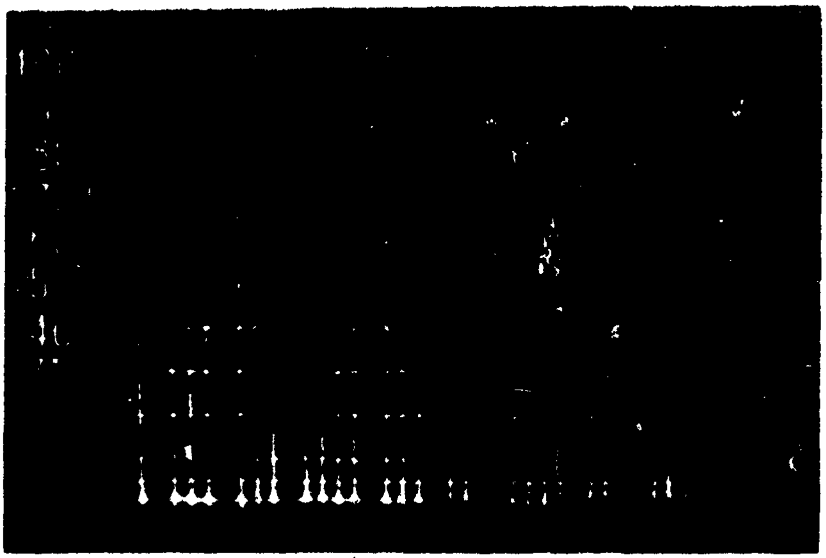


Figure 48. Example of the depiction of the spectrum of a noise on the screen of a spectrometer.

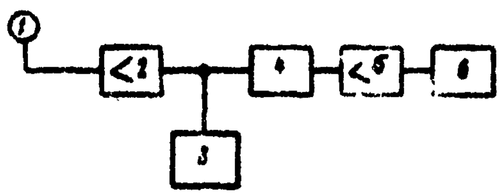


Figure 49. Block diagram of a very simple heterodyne analyzer.

(1) Electroacoustic converter; (2) amplifier; (3) heterodyne; (4) filter; (5) amplifier; (6) registering instrument.

An example of the depiction of the frequency curve of the noise of a machine obtained with the aid of a narrow-band analyzer is shown in Figure 50. Comparison with computations indicates that in the given case the peaks at frequencies of 50 and 100 cycles are due to the presence of a disbalance in rotating mass in the machine, and the peaks at frequencies of 640 and 1100 cycles are caused by the operation of gears in the machine.

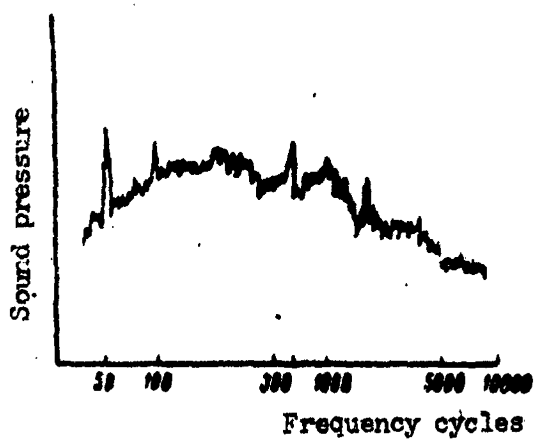


Figure 50. **Example of spectrogram obtained with the aid of narrow-band analysis of noise.**

To ensure an undistorted picture the speed of change of frequency of the signal of the heterodyne must not exceed a certain value

$$v = \frac{(\Delta f)^2}{4}, \text{ where}$$

Δf is the band of admission of the analyzer.

The width of the band of admission of the analyzer determines its resolving power, i.e., its ability to separate closely adjacent frequency components of sound or vibration. In contemporary practice acoustic measurements often make use of analyzers with 4 and 20-cycle band widths.

The time for analysis of a frequency band $f_1 - f_2$ by an analyzer with constant width of band of admission is equal to:

$$t = \frac{f_2 - f_1}{v} = 4\,\frac{f_2 - f_1}{(\Delta f)^2}\,. \tag{88}$$

If $f_2 \gg f_1$, then

$$t \cong \frac{4f_2}{(\Delta f)^2}\,. \tag{89}$$

With a sufficiently narrow band of admission the time of analysis may entail several tens of minutes.

During recent years a design of analyzer with constant relative width of admission γ has found ever increasing use. This value γ is equal to:

$$\gamma = \frac{\Delta f}{f_{av}} = \text{const}; \tag{90}$$

where f_{av} is the average frequency of the band of admission.

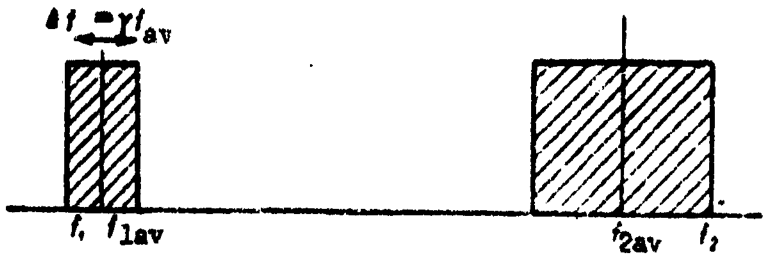


Figure 51. Determining the time for analysis of a sound using an analyzer with constant relative width of the admission band.

In this case the absolute width of the admission band is variable. For example, at $\gamma = 0.1$ the width of the band of admission at a frequency of 100 cycles is $\Delta f = 0.1 \cdot 100 = 10$ cycles, but at a frequency of 5,000 cycles Δf is equal to 500 cycles. Broadening the band of admission by increasing the frequency not only decreases the analysis time, but (which is most important) gives the frequency scale of the analyzer a logarithmic character corresponding to the subjective perception of sounds of different tone. Furthermore, changing the band of admission of the analyzer with respect to frequency enables avoidance of distortion of the recording of the components of high frequencies (cf. Chapter 20). In this, however, the resolving power of the analyzer is diminished by a definite amount at high frequencies.

Let us evaluate the time necessary for the analysis of a noise, using an analyzer with constant relative admission band. On the basis of formulas (88) and (90), this time is equal to:

$$t_1 = \frac{4}{f_2 - f_1} \int_{f_{1av}}^{f_{2av}} \frac{f_2 - f_1}{(\Delta f)^2}\, df_{av} = 4 \int_{f_1\left(1+\frac{\gamma}{2}\right)}^{f_2\left(1-\frac{\gamma}{2}\right)} \frac{df_{av}}{(\gamma f_{av})^2}. \tag{91}$$

The expression of the limits of the integration f_{2av} and f_{1av} by means of the corresponding values f_2, f_1 and γ is explained by Figure 51, where the shaded areas correspond to the bands of admission of the analyzer at the lower and upper frequencies of analysis.

After integration we obtain:

$$t_1 = \frac{4}{\gamma^2} \frac{f_2\left(1-\frac{\gamma}{2}\right) - f_1\left(1+\frac{\gamma}{2}\right)}{f_2 f_1\left(1-\frac{\gamma^2}{4}\right)}.$$

Because usually $f_2 \gg f_1$, and $\gamma \ll 1$, we have

$$t_1 \cong \frac{4}{\gamma^2 f_1}. \qquad (92)$$

Comparison of formulas (89) and (92) indicates that while in the case of an analyzer with constant absolute width of the band of admission, the analysis time is determined mainly by the higher frequencies of the analysis, the time of analysis with an analyzer with constant relative width of admission is determined by the lower frequencies of the analysis.

Example 21. The problem is to determine the ratio between the time of undistorted analysis of a noise in a frequency range of 50 to 5,000 cycles using an analyzer with an admission band of $\Delta f = 10$ cycles, and the time of analysis using an analyzer with a relative width of admission band $\gamma = 0.1$.

Solution. From formulas (89) and (92) the ratio of analysis times is:

$$\frac{t}{t_1} \cong \left(\frac{\gamma}{\Delta f}\right)^2 f_1 \cdot f_2 = \left(\frac{0{,}1}{10}\right)^2 \cdot 50 \cdot 5000 = 25.$$

The absolute time of analysis with an analyzer having a constant admission band (formula 89) is

$$t \cong \frac{4 \cdot 5000}{(10)^2} = 200 \quad \text{sec.}$$

In the case of an analyzer with constant relative width of the band of admission, the analysis time in the given case does not exceed t = 8 seconds.

With reduction of Δ f to 4 cycles and γ to 0.04 the values of the analysis times t and t_1 increase correspondingly to 21 minutes, and 50 seconds.

We may mention once again another method of recording sounds and noises, visualization. The visualization method of depicting a sound was originally proposed for the recording of speech. The recording is made on a photographic film, in which the horizontal axis represents time and the vertical axis the frequencies of the sound components, and the degree of darkening of the individual portions of the film is a measure of the intensity of the corresponding components of the sound at a given moment of time.

The cathode spectrometer may be recommended as an analytic and recording instrument. In distinction from the usual systems of application of the spectrometer, the output signal of the amplifier is connected not with the controlling plate of the cathode tube, but with a device modulating the clarity of illumination of the tube. On the screen of the tube are, instead of a series of lines corresponding to the amplitudes of the sound components in various frequency bands, appears a series of points with varying intensity of illumination. A motion picture camera with an uninterrupted length of film is set up in front of the screen.

Figure 52 shows spectrograms obtained by the method described by the author, using a spectrometer with one-third-octave filters (53). The first spectrogram (Figure 52-a) shows the record of the signal of a tube-type noise generator. As is apparent, the generator gives a broad range of frequencies of vibration. In the given case the range of operation of the series of filters is 50 to 20,000 cycles.

Figure 52-b shows the spectrum of a noise generated by starting and stopping an internal combustion engine. Figure 52-c shows the spectrogram of the sound of a whistle. The high frequency components are clearly expressed (in distinction from Figure 48, the frequency scale in this case is vertical).

If the photographs of Figure 52 comprise depiction of the spectrum picture of a noise according to time, Figure 53 shows the depiction of the same picture in space. The record of noise shown was taken while moving a microphone along a passageway in a ship. The figure shows how the noise increases in strength at the location of hatches to the machinery spaces, where auxiliary Diesel generators are in operation.

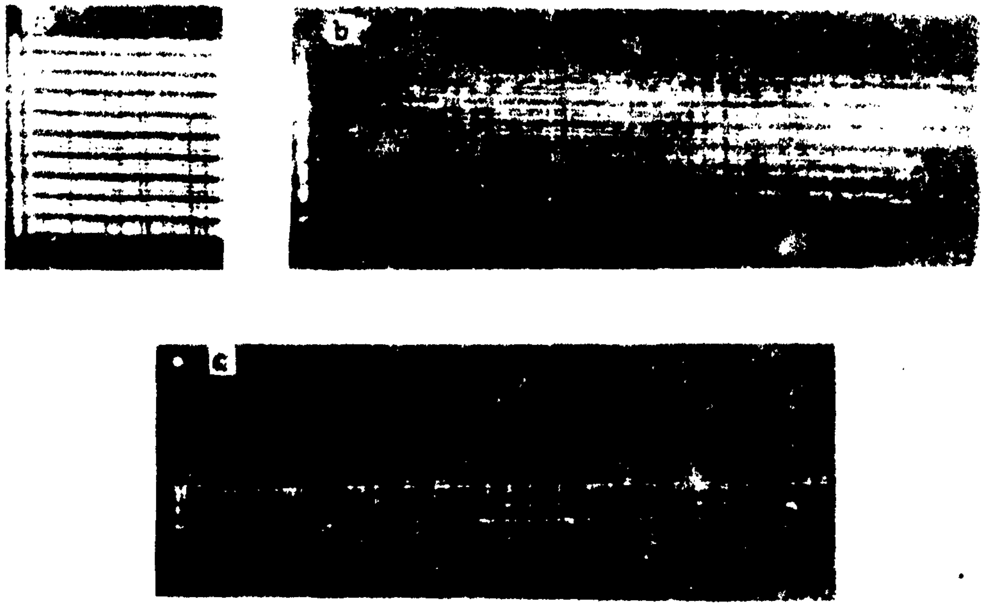


Figure 52. Depiction of the spectrum of noises according to time, obtained by the method of modulation of the brightness of illumination of the tube of the spectrometer.

(a) Signal of a noise generator
(b) Starting and stopping a Diesel engine
(c) Sound of a whistle.

When ordinary motion picture film is used the dynamic range of photographic records of this type does not exceed 15 db. This range may be expanded with the use of phototelegraphy. In the interpretation of the recordings, the density of darkening of individual portions of the film is determined with the aid of a photometer or densitometer.

The visualization method presents a visual representation of the character of the changes of noise or vibration during changes in the operating regime of machinery, and enables a picture of the spectral distribution of the noise in a compartment to be obtained, plus evaluation of the effects of various sound-protection structures. It may be suggested that this method be applied more extensively in the practice of acoustic measurement.

Passageway

Figure 53. Spectral record of noise encountered in moving a microphone along a passageway (hatchways to the machinery spaces open)

18. <u>Expressing of the Results of Analysis of Sound and Vibration.</u>

In the construction of the spectrogram of a noise or vibration the values of the boundaries of the frequency band or the average frequencies of these bands, or number of the bands are plotted along the horizontal axis, and the noise levels in these bands are recorded on the vertical axis. Inasmuch as the scale of frequencies conforms to the logarithmic law (cf. Paragraph 6), the average frequencies of the band are not the arithmetic means, but the geometric means of the values of the boundary frequencies of the band. For a frequency band equal to one octave, the mean frequency is found by means of the expression:

$$f_{av} = \sqrt{f_u \cdot f_\ell} = \sqrt{2f_\ell \cdot f_\ell} = 1,41 f_\ell. \qquad (93)$$

where f_u and f_ℓ are the upper and lower boundaries of the frequency of the octave.

In like manner we have for half-octaves:

$$f_{av} = \sqrt[4]{2}\, f_\ell = 1,19 f_\ell \qquad (94)$$

and for third-octaves:

$$f_{av} = \sqrt[6]{2}\, f_\ell = 1.13 f_\ell \qquad (95)$$

Occasionally, as in comparison of the levels of spectrograms which have been obtained with different widths of the band of admission, it becomes necessary to divide octaves into half-, and third-octaves. It is not difficult to see that an octave is divided into half-octaves by its geometric-mean frequency, i.e., if the limiting frequencies of any given octave are equal to f_{ℓ} and f_u, the limiting frequencies of the two half-octaves of this octave are:

$$\left.\begin{array}{l|l} \text{first half-octave} & f_{\ell} - 1{,}41 f_{\ell}; \\ \text{second half-octave} & 1{,}41 f_{\ell} - 2f_{\ell} = f_u. \end{array}\right\} \qquad (96)$$

The frequencies dividing the octave f_{ℓ} -- f_u into third-octaves are:

$$\left.\begin{array}{l|l} \text{first third-octave} & f_{\ell} - \sqrt[3]{2}\, f_{\ell} = 1{,}26 f_{\ell}; \\ \text{second third-octave} & 1{,}26 f_{\ell} - \sqrt[3]{4}\, f_{\ell} = 1{,}59 f_{\ell}; \\ \text{third third-octave} & 1{,}59 f_{\ell} - 2f_{\ell} = f_u. \end{array}\right\} \qquad (97)$$

Although the method of depiction of spectrums in bands equal to an octave or parts of an octave, is widespread, it is more rational from the point of view of unification to express the results of analysis in terms of a uniform width of frequency band, equal to 1 cycle. Levels in the 1-cycle band are called spectral levels. When the levels in the bands of finite width are brought to the spectral levels, it follows from the foregoing that the level does not change with frequency within the limits of the given band (cf. Paragraph 5). Thus the intensity of sound, reduced to 1 cycle, is equal to:

$$J_{sp} = \frac{J_b}{f_u - f_{\ell}},$$

where J_b is the intensity of sound in the given band of frequency f_{ℓ} -- f_u.

Relating the intensity in both parts of the equation to the threshold intensity and converting to logarithms, we obtain the expression for the spectral level in the following form:

$$\beta_{sp} = \beta_b - 10 \log (f_u - f_{\ell}) \text{ db.} \qquad (98)$$

Here β_b is the level of the strength of the sound in the frequency band (in decibels) above threshold.

If the analyzing apparatus has a constant relative band of admission γ, then

$$f_u - f_l = f_{av} \cdot \gamma .$$

For this type of analysing instrument the spectral level at frequency f_{av} is equal to:

$$\beta_{sp} = \beta_b - 10 \log (f_{av} \cdot \gamma) \text{ db.} \qquad (99)$$

Formulas (98) and (99) also enable resolution of the opposite task: finding β_b, the level at any required frequency band according to the value of the spectral level.

Figure 54 shows a graph for converting the levels of frequency bands of various width into spectral levels, and vice versa; the corresponding correction, determined from the vertical axis of the graph, is taken as plus or minus.

Example 22. A sound level in the half-octave 1,000 - 1,410 cycles is equal to 80 db above threshold. The problem is to determine the magnitude of the spectral level and the level measured at the average frequency of the half-octave by an analyser with a 3-percent relative width of band of admission.

Solution. The average frequency of the half-octave (formula 94) in the given case is equal to:

$$f_{av} = 1.19 \cdot 1{,}000 = 1{,}190 \text{ cycles.}$$

At this frequency, according to Figure 54 the correction for bringing the level of the half-octave to a level in a 1-cycle band is 26 db. Thus the spectral level is equal to 80 - 26 = 54 db.

At this frequency the level measured in the 3-percent band exceeds the spectral level by approximately 16 db (cf. the broken curve in the middle set of curves relating to the 100 to 10,000-cycle range of frequencies). The level measured with a narrow-band analyser is equal to 54 + 16 = 70 db, i.e. 10 db lower than the half-octave level.

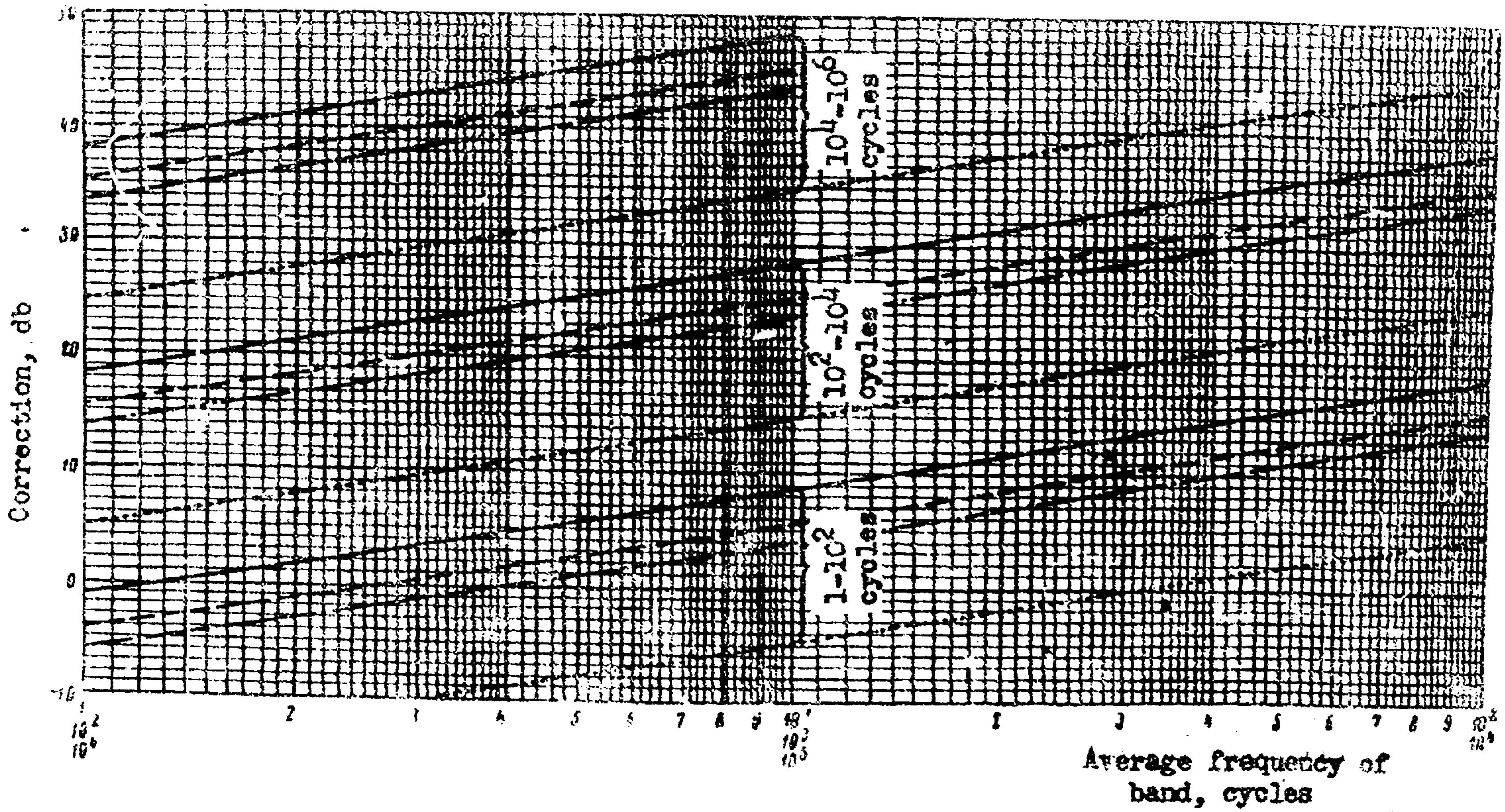


Figure 54. Correction for converting levels expressed in frequency bands of various widths into spectral level.

——— With of band 1 octave; - - - - - Width of band 1/2 octave;

-·-·-·-· Width of band 1/3 octave; ······ Width of band 3 percent.

Example 23. The spectrum of vibrations of a machine is such that the spectral level of vibratory acceleration $\beta_{sp.a}$ is equal to 60 db for the entire range of investigated frequencies (the straight line in Figure 55). The problem is to plot the spectral level of the velocity of vibration and the octave levels of the velocity of vibration and vibratory acceleration as a function of frequency, assuming that these levels at a frequency of 1,000 cycles coincide.

Solution. From formula (98) we find the octave level of vibratory acceleration:

$$\beta_{O.a} = \beta_{av.a} + 10 \log (f_u - f_{\ell}) \text{ db.}$$

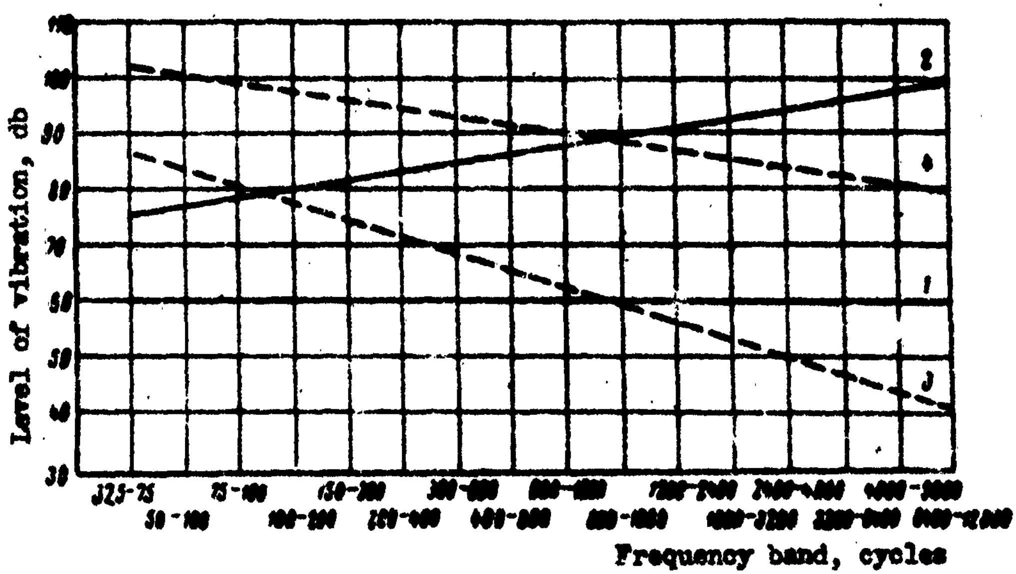


Figure 55. Vibration levels as a function of frequency (Example 23).

1 - Spectral level of vibratory acceleration; 2 - octave-band level of vibratory acceleration; 3 - spectral level of speed of vibration; 4 - octave-band level of speed of vibration.

The values of the boundary frequencies of the octaves frequently used in practice are marked on the horizontal axis of the graph (figure 55). For the purpose of increasing the number of points of reference the octaves are overlapped. Let us compute the level $\beta_{0.a}$ for two octaves, fairly well separated from each other: 50 - 100 cycles, and 3,200 - 6,400 cycles.

The level of acceleration in the first of the selected octaves is:

$$\beta_{0.a_1} = 60 + 10 \log (100 - 50) = 77 \text{ db.}$$

The level of acceleration for the second octave is:

$$\beta_{0.a_2} = 60 + 10 \log (6{,}400 - 3{,}200) = 95.5 \text{ db.}$$

The same level values may be found, taking the corrections for the average frequencies of the octaves from Figure 54, which are equal to:

$$f_{av_1} = 1.41 \cdot 50 = 71 \text{ cycles;}$$
$$f_{av_2} = 1.41 \cdot 3{,}200 = 4{,}500 \text{ cycles.}$$

The correction according to the graph is used with a plus sign, because the reciprocal problem is resolved in the present case: the level in a wide frequency band is found according to the spectral level.

The computed level values are plotted against the average frequency of the octaves, and the straight line 2 is drawn through the points thus obtained (the levels as a function of frequency in the given case form straight lines, because the frequencies and levels are plotted on a logarithmic scale).

In finding the level of the velocity of vibration we use, as mentioned in Paragraph 10, the fact that the value of the speed of vibration of air particles at the threshold of hearing, equal to $\dot{y}_0 = 5 \cdot 10^{-6}$ cm/sec, may be taken as the zero value of the speed of vibration of the oscillating surface. This threshold speed also was used in the starting conditions of the present example. From the condition of equality of the levels of speed and acceleration at a frequency of 1,000 cycles it follows that:

$$\frac{\dot{y}}{5\cdot 10^{-6}} = \frac{2\pi\cdot 1000\dot{y}}{\ddot{y}_0}.$$

From this we find the numerical value of the threshold of vibratory acceleration:

$$\ddot{y}_0 = \pi\cdot 10^{-2} \text{ cm/sec}^2.$$

At any given frequency f the difference between the spectral levels of vibratory acceleration β_{sp-a} and the speed of vibration β_{sp-v} is equal to:

$$\beta_{sp-a} - \beta_{sp-v} = 20 \log \frac{2\pi f\dot{y}}{\pi \cdot 10^{-2}} - 20 \log \frac{\dot{y}}{5 \cdot 10^{-6}} =$$

$$= 20 \log (10^{-3}f) = 20 \log f - 60 \text{ db.}$$

If the average values of the frequency of the two octaves chosen in the above are inserted in the last previous formula, we obtain, for the octave 50 - 100 cycles:

$$\beta_{sp-a} - \beta_{sp-v} = -23 \text{ db};$$

and for the octave 3,200 - 6,400 cycles:

$$\beta_{sp-a} - \beta_{sp-v} = 13 \text{ db.}$$

Plotting the values of these differences at the average frequencies of the octaves from the ordinate of the straight line 1 and joining the obtained points, we obtain straight line 3, characterizing the spectral level of the velocity of vibration.

The octave-band level of the velocity of vibration (straight line 4), as is the case with acceleration, also, exceeds the spectral level at the average frequencies of the octaves by 17 and 35.5 db, respectively.

The graph obtained in the above visually reveals the inaccuracy of characterizing vibration with the term "vibratory spectrum" without supplementary clarification. A spectral characteristic for one and the same vibration may be obtained either increasing with frequency, decreasing in frequency or independent of frequency, depending upon the fixed parameter of vibration and the width of the frequency interval in

which the vibration is measured. Also, the difference in the levels of vibration of the characteristics at one and the same frequency may attain several tens of decibels.

As a rule, the spectral levels and levels in any given frequency band are plotted in decibels above the zero threshold of noise or vibration. In the case in which the measuring apparatus lacks absolute calibration or comparative measurements are being performed, the levels in individual frequency bands may be plotted relative to the maximum level of the spectrum. In several models of analyzers the amplitude scale is constructed in a way so that 100-percent amplitude corresponds to the maximum component of the spectrum.

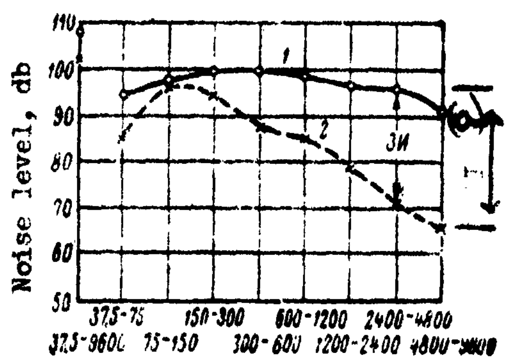


Figure 56. Frequency curves of the noise levels of a machine.

1 - Without sound absorbing structure;
2 - With sound absorbing structure
a - Acoustic effect of structure, db (difference between curves 1 and 2)

In comparing the levels of noise in different rooms or deriving from different sources, and in determining the value of sound isolation or the sound absorption of any given device, a series of spectral curves are plotted on a graph. Let us assume that the upper curve depicts the spectrum of noise in a room prior to installation of a noise elimination structure housing, and the lower curve represents the level after installation. Then the difference in the ordinate of the curves at each frequency determines the acoustic effect of the structure, ZI, expressed in decibels. In similar comparative measurements and evaluations the ordinates of the curves may be plotted from any given starting level (not necessarily standard).

The levels of the entire band of frequencies measured are usually plotted on the axis (in Figure 56, for example, this band extends from 37.5 to 9,600 cycles). The total level exceeds the maximum component of the spectrum or the level in any of its frequency bands by 3 to 10 db, in which the first number refers to the spectral curve with a markedly outstanding value in one of the bands (lower curve in Figure 56), and the second refers to a rather flat spectral curve (upper curve).

[Note: Although the translation is correct, the author seems to have made an inadvertance here. Note that the upper curve in Figure 56 is rather flat. This part of the sentence is technically incorrect as originally written (according to the principles given earlier for the addition of decibels to obtain total levels).]

CHAPTER 5

NOISES IN SHIPS

19. Sources of Noise, the Paths by Which It Is Propagated in Ships

The main sources of noise in ships are the main and auxiliary machinery. Very unpleasant noises may arise in the ventilation systems for the ship and the machinery spaces, in systems for pumping liquids, in air conditioning apparatus, and service equipment (elevators, sanitation units), etc.

Occasionally another source of noise is encountered, which may be called a secondary source. We mean by this noises arising from the rattling together of inadequately fastened deck plating, pipes and other metallic parts and structures due to the action of the vibration which takes place in ships under way. This vibration occurs at infrasonic frequencies, although its consequences appear in the form of unpleasant clanking, rattling and jarring.

No less irritating to ship passengers are noises connected with the presence on the ship of other passengers, such as loud voices and laughter, shouts of children, footfalls in the passageways and on deck. The continuous loud playing of radios also has been established as having an irritating effect on resting passengers.

The above described sources of sounds and noises are located within ships. In fairly rare cases the source of unpleasant noises in a certain portion of a ship's compartment may consist of the propeller, especially if the propeller is inclined to "sing," i.e., production of a tonal sound by movement of the blades. In the stern sections of high speed ships, low frequency sounds may be heard resulting from the vibration of hull plating due to the periodic sucking action of the propellers and the noise of the propellers, themselves, caused by vibrations and the banging of air and vapor cavities on the propeller blades.

The types of propagation of sound in ships include the following (Figure 57):

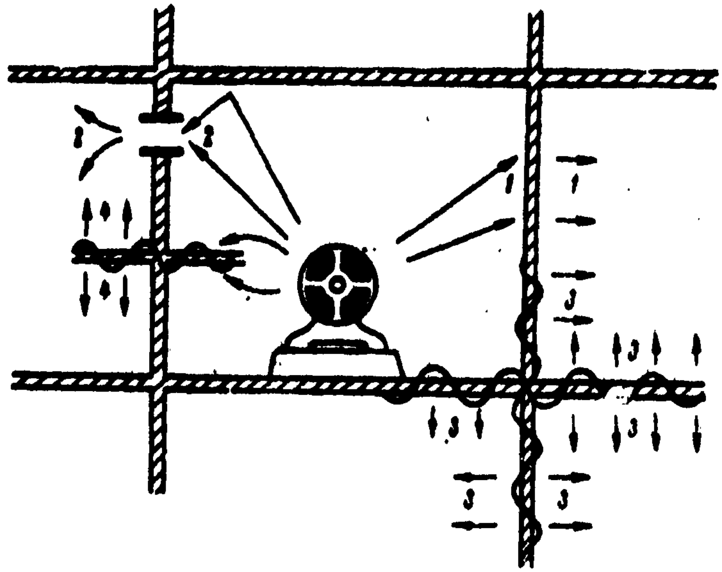


Figure 57. Ways in which sound vibrations are transmitted from the source of the sound into neighboring compartments.

1 - Through the air and through partitions;
2 - Through openings in partitions;
3 - Through the flooring and structural parts;
4 - By means of the excitation plates passing through insulating structures.

(1) Transmission of airborne sound radiated by the source, through walls, overhead, bulkheads and the deck; at low frequencies this transmission occurs as a result of membrane-vibrations of the partitioning structures, and at high frequencies it has a wave character.

(2) Transmission of airborne sound through openings, hatches, ventillation conduits, slots, and non-tight door casings;

(3) Transmission of sonic vibrations through foundation and hull structures, with subsequent radiation of airborne sound in the neighboring and in remote compartments.

Another type of transmission of sound is possible, which apparently was indicated for the first time by Gesele (124), consisting of excitation of airborne sound by bending vibrations in broad expanses of plating.

If the plate extends into the neighboring compartment, its vibration evokes radiation of sound in this room. Only a small portion of the energy may be transmitted in this way, although even this amount of energy may considerably increase the noise level in neighboring compartments in the case of high sound isolation of bulkheads and partitions.

The relationship between the levels of noise transmitted in various ways depends upon the character of the source of the noise and the structural design of the boundaries of the compartment. The first two paths of sound transmission are very important in the transmission of sound to neighboring compartments in the case of machinery with a high air noise level (ventillators) installed in compartments with light partitioning, or partitioning with a large number of holes and openings. In the case of machinery in which the vibratory energy is produced mainly in the form of sonic vibration (pumps and compressors), noise in neighboring and remote compartments may be due mainly to the third type of sound transmission. This is true particularly for non-isolated machines mounted on relatively light foundations in compartments of which the bulkheads isolate airborne noise very well. Noise in ship compartments remote from the source of vibration almost always may be explained by the transmission of sonic vibrations through the structure of the hull.

20. Noise and Vibrations in Ships

In the following are presented tables and graphs of noise level in several ships (according to the measurement data of L. G. Vasil'yev, A. P. Dyatchenko, V. I. Zinchenko, V. F. Lyusov, V. M. Spiridonov and the present author).

Table 5 presents the noise level in compartments of the Soviet-built Diesel-electric ship "Kurgan."* The habited areas of this ship are provided with standard heat insulation. No special measures have been taken in this ship with respect to noise reduction.

As is apparent from the table the level attains considerable magnitude in various compartments. In the engine room the level is 115 to 118 db, but in the vicinity of the engine's air blower it exceeds

*[Note: The "Kurgan" is a refrigerated cargo vessel, built about 1957, of 10,460 tons full load displacement. The main propulsion plant consists of 4 diesel generator sets. Each diesel is Russian Model 3D100, which is a copy of the Fairbanks-Morse diesel Model 38D8 1/8.]

the threshold of pain perception. The considerable noise at the site of the telephone installation practically precludes the possibility of use of the telephone. In the sick bay, the doctor's cabin and several other cabins the noise level attains 81 to 83 db.

TABLE 5

NOISE LEVELS IN COMPARTMENTS OF A DIESEL-ELECTRIC SHIP

Site of Noise Measurement	Noise level, in db, with number of operating diesels (810 rpm)		
	2	3	4
Forward engine room between Diesel generators	117	118	118
Entrance to central control station	115	117	118
Central control station (door to engine room closed)	94	96	98
Same, with door open	—	—	107
Point 0.6 m from air blower of Main Diesel Generator No. 1	—	—	121
Telephone station in engine room	—	—	117
Upper grating of ventillation shaft (at level of boat deck)	115	116	116
Passageway at door to engine room	93	95	95
3rd Engineer's cabin	—	—	83
Passenger cabin No. 25	74	74	76
Passageway at cabin No. 29	88	91	93
Lazarette	81	81	83
Dispensary	79	80	84
Doctor's cabin	81	83	82
Red corner (Communists') club room	—	—	76
Hold of aft engine room at main electric motor	102	104	106
Same, at aft bearing of propeller shaft	100	101	103
Hold of forward engine room, under main diesels	108	110	110
Aft engine room, near auxiliary engine No. 2	108	109	109

The spectral curves of noise (Figure 58) indicate that the maximum sound level in the engine room is observed in the range of the 3rd to 5th octaves. Outside the engine room the greatest level in the noise spectrum is observed at the lower frequencies. This results from the fact that low frequency vibrations are weakly isolated and damped with respect to propagation through the hull structure.

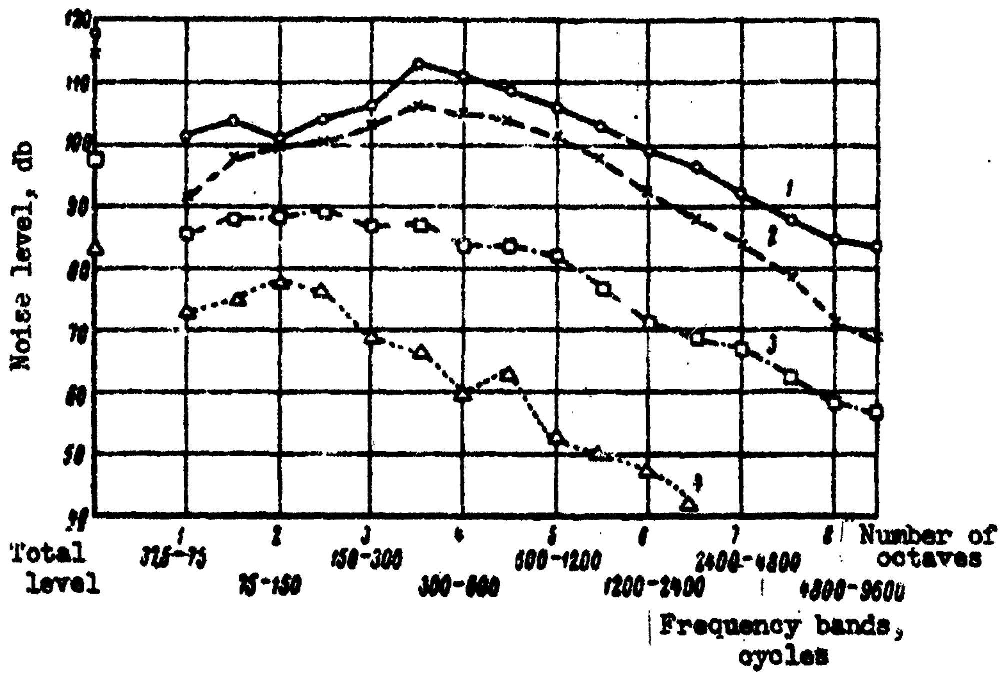


Figure 58. Spectra and levels of noise in compartments of the Diesel-electric ship "Kurgan."

1 - In engine room, between Main Diesel Generators Nos. 2 and 3; 2 - at the upper grating of ventillation shaft; 3 - at the central control station (door to engine room closed); 4 - in the dispensary.

Figure 59 shows the noise level as a function of ship's speed for a Diesel ship. With a change from slow to full speed, the noise levels in the various compartments change by 4 or 5 db. When the ship is stopped and only the auxiliary Diesel generators are operating, the noise level still is high and surpasses the level of the main propulsion engines at slow speed.

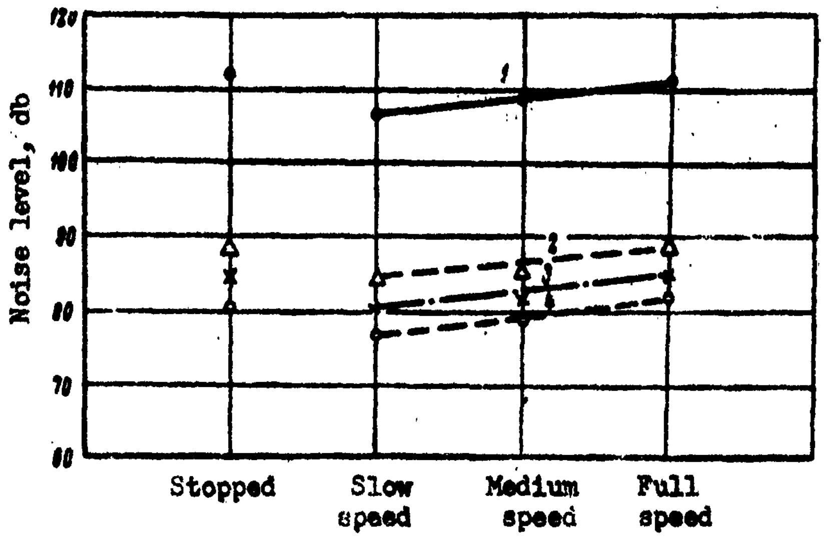


Figure 59. Noise levels on a Diesel ship

1 - In the engine room; 2 - in the cabin of the 2nd engineer; 3 - in the first crew's quarters; 4 - in the second crew's quarters.

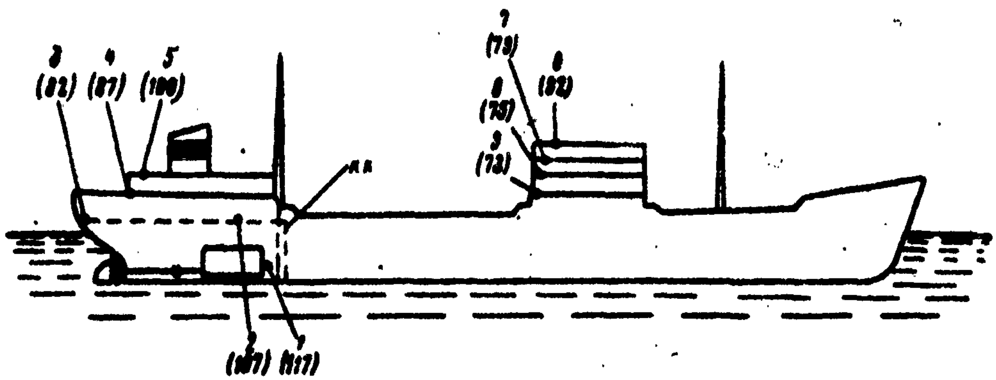


Figure 60. Noise levels (figures in parentheses) in the engine room, on decks and bridges of a tanker.

1 - In the engine room; 2 - on aft deck above the machinery installation; 3 - at aft end of upper deck; 4 - at aft end of quarterdeck house; 5 - on quarterdeck house roof; 6 - at open portion of lower bridge; 7 - at lower bridge; 8 - on the boat deck; 9 - on the spar deck; KK - cofferdam.

The picture of noise levels on a tanker (Figure 60) indicates that the greatest levels, as in other ships, are observed in the engine room. On the bridges and the amidships superstructure decks, separated from the engine room by a cofferdam and extensive hull structures, the noise is 30 to 40 db lower than in the engine room.

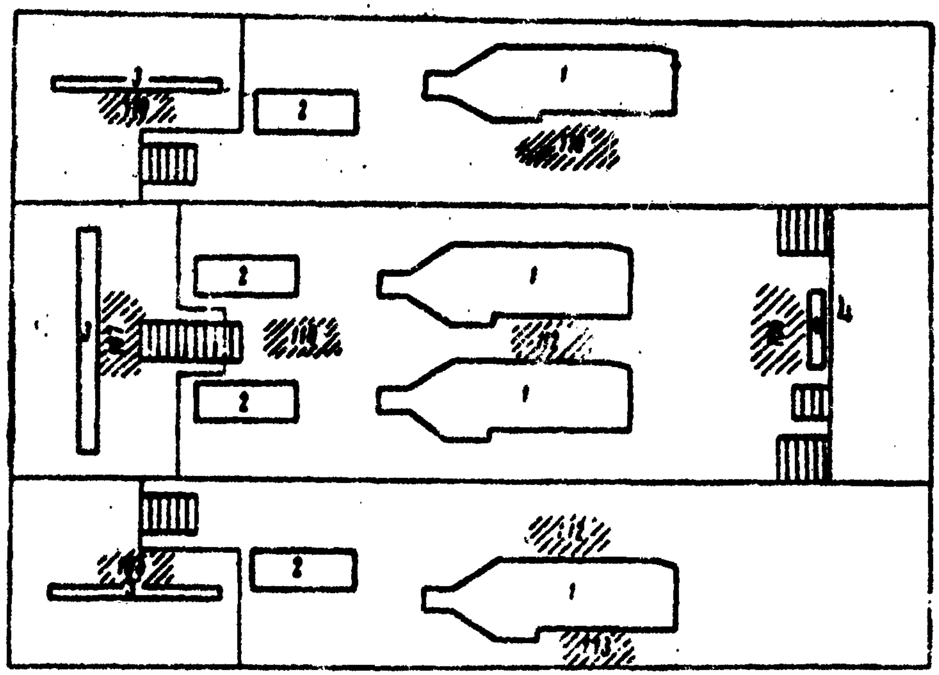


Figure 61. Noise levels in engine rooms of a Diesel-electric ferry boat.

1 - Main engines; 2 - auxiliary engines; 3 - control panels; 4 - control station.

Figure 61 presents a general diagram of the engine room of the Diesel-electric ferry boat "Yuzhnyy" ["Southern"] (of the Krymsko-Kavkazskaya [Crimean-Caucasus] Line). The engine room encloses a space of 1,500 m^3 and is subdivided by longitudinal bulkheads into three parts. In these compartments are located four main Diesels (model D50*) and four auxiliary Diesels (7D6). The motors occupy a relatively

*[Note: The diesel engine D50 is a copy of the American Locomotive Company's engine Model 6-12 1/2 x 13-T (with exhaust gas turbo blower). - Translator's note.]

small portion of the area of the engine room, and thus there are extensive possibilities for the introduction of sound isolating and sound absorption structures. However, these structures are lacking, as a result of which high noise levels are observed both near the Diesels (110 to 113 db) and at the control panels of the control station (105 to 110 db)

TABLE 6

NOISE LEVELS OF VARIOUS SHIPBOARD MACHINERY

Machinery	Noise Level, db	Place of Measurement
Main Propulsion		
Main Diesel, MAN, 3,000 hp, 250 rpm, (five engines in operation)	106	Diesel-electric Ship "Rossiya"
Auxiliary Diesel 7D6, 150 hp, 1,500 rpm	115	Diesel-electric ship "General Azi Aslanov"
Main Diesel 3D6, 150 hp 1,500 rpm	115	River passenger launch "Moskvich"
Main Diesels "Burmeister and Wain" 5,420 hp, 125 rpm (two engines in operation	96	Motor Ship "Gruziya"
General Ele[illegible]ic steam turbine, 2,500 hp	92	Turbine Ship "Omsk"
Compound steam engine, 350 hp, 200 rpm	78	Steamship "Rota"
Compressors		
Diesel compressor 1K	110	Test stand
Electric compressor EK-15, 550 rpm	96	Test stand
Refrigeration Equipment		
Refrigerator E-250 (output 250,000 cal/hr)	80	Test stand
Refrigerator PAM-FV2 (output 2,000 cal/hr	80	Test stand

The adduced data relate mainly to main propulsion Diesel power plants. Table 6 shows, in addition to the noise of Diesels, analogous levels of some other ship equipment. It is apparent that steam turbine

power plants, and especially steam engines, have considerably less noise. Diesel plants of low rpm, even the more powerful ones, are less noisy than high-speed Diesels (cf. Chapter 17).

As a rule, Diesel compressors are noisier than electric compressors powered from the ship's electrical system, which urges their preference on passenger ships. Irrespective of their power, the noise of refrigeration machines is on the order of 80 db, i.e., less than the noise of the engines.

TABLE 7

NOISE LEVELS IN COMPARTMENTS OF A PUSHER TUG, FOR VARIOUS METHODS OF SUPPORTING THE DIESEL GENERATORS

Compartment	Noise Level, db, With Operation of:	
	Left Diesel Generator (mounted on stiff rubber vibration isolation mounts)	Right Diesel Generator (mounted on very soft spring vibration isolation mounts)
Diesel generator room	105	106
Saloon	85	74
Cabin of 1st engineer	93	79
Cabin of 2nd engineer	88	72
Cabin of 3rd engineer	88	74
Cabin of Chief Engineer	83	71
Cabin of captain	75	72
Cabin of 2nd helmsman	87	75
Radio room	81	71
Pilot house	81	69

Table 7 presents data on noise levels aboard a powerful river pusher tug. The data of this table indicate that, on small ships, noise in all compartments except the engine room has its origin in the sonic vibrations of the hull structure. Actually, the noise in all compartments is reduced markedly by increasing the vibration isolation properties of engine mountings, although the airborne noise of the engine remains unchanged.

The levels of acceleration of sonic vibration in a Diesel ship, measured by a piezoelectric vibrometer, are shown in Figure 62.

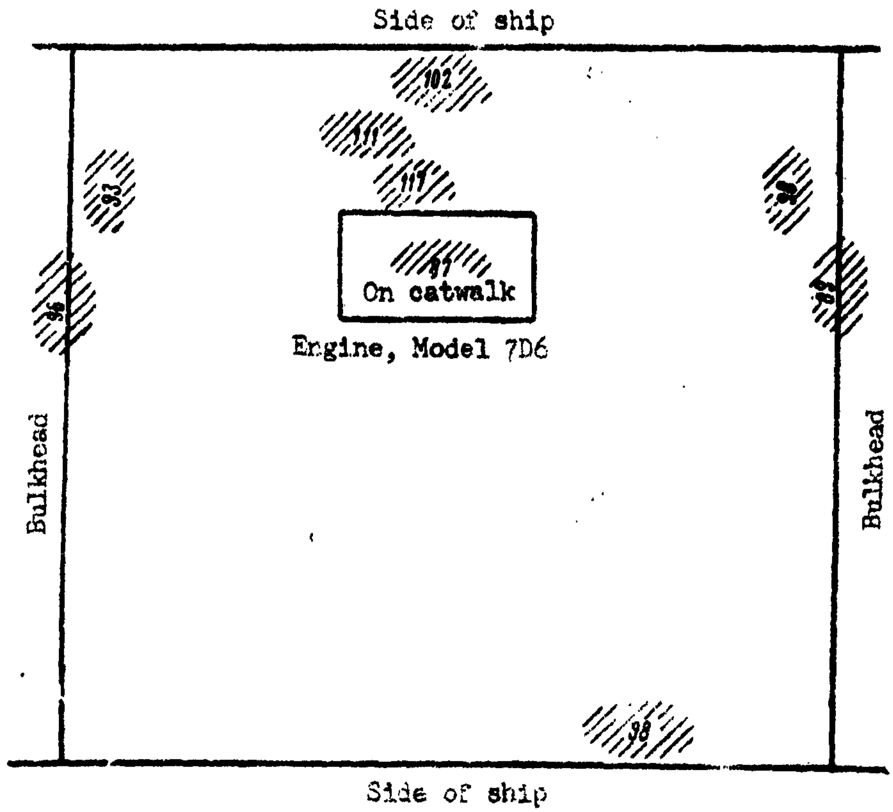


Figure 62. Sonic vibration levels in a Diesel ship.

21. Standards for Noise

Detailed norms of noise in shipboard compartments are lacking at the present time. The shipbuilding regulations issued by the Maritime Registry of the USSR (86) indicate that the noise level in the engine room of ships must not exceed 100 db at a distance of 1 meter from any item of machinery. At the same time, the levels of the components of the noise in the frequency range above 700 cycles must be less than in the lower frequencies.

In all cases audibility of sound signals from the captain's bridge to the control station must be ensured at all points of the engine room. If the indicated norms for noise of machinery and service personnel are exceeded, sound isolating structures must be installed.

The noise level must not exceed 80 to 90 db in work areas, and the noise must satisfy the above-mentioned requirement with respect to its frequency composition.

The Gossaninspektsiya [State Sanitation and Hygiene Inspectorate] has developed more complete norms of noisiness (I. I. Slavin), based on criteria of the physiological harmfulness of noise. Noise is broken down into several groups on the basis of magnitude of level in these norms. The higher the noise level, the lower is the permissible limit of high frequency components. The requirements may be illustrated by a graph (Figure 63), representing the permissible levels of noise in the various octaves as a function of frequency. The straight line connecting these permissible levels passes through a level of 100 db in the first octave and drops by approximately 4 db per octave as frequency increases. The noise level may not lie above this straight line in any of the octaves.

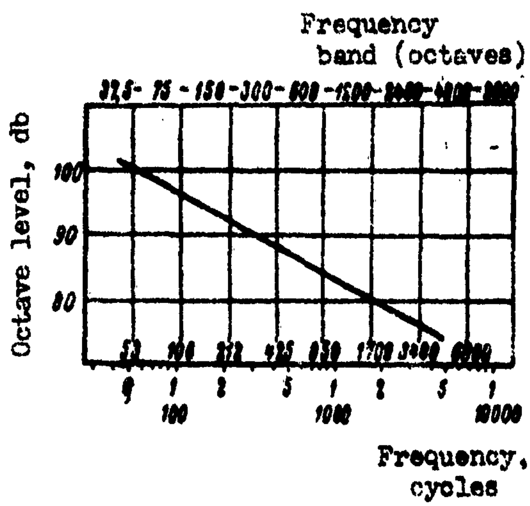


Figure 63. Frequency curve of the permissible noise levels.

It is recommended that a noise level not exceeding 80 to 90 db be permitted in ordinary production work, and not exceeding 60 db be permitted in quiet production work. The noise level must not surpass 30 to 40 db in habited areas in view of the requirement pertaining to absence of harmful conditions and the demands of comfort (with respect to ships this figure may be increased to 40 or 50 db).

Comparison of the noise level values presented in the preceding paragraph with the permissible levels shows that the noise level in the engine room surpasses both the norms of the Registry and the norms of Gossaninspektsiya. The Registry requirement relative to ensuring audibility of signals from the bridge have not been fulfilled because the intelligibility of signals is negligible at the noise level of 105 to 110 db at the control stations. In this connection measures are necessary for reduction of noise in compartments of Diesel ships, especially in the engine room.

22. Problems, Methods and Means for Controlling Noise in Ships

The following two practical problems of shipboard acoustics connected with reduction of noise may be mentioned: reduction of the unpleasantness and loudness of noise; increasing the intelligibility of speech.

Resolution of the first problem requires primarily a decrease in the unpleasant high frequency sound vibrations. This problem is relatively simple, because almost all sound-protection devices are effective at high frequencies.

It is necessary that sufficient intelligibility of speech be ensured for understanding commands, and to permit telephone conversations of the ship's personnel and passengers. This is a more complex task because it requires reduction of noise in the range of the main components of speech (300 to 3,000 cycles), and to the extent possible, in the lower band of frequencies. Electroacoustic methods also are used to increase the intelligibility of telephone conversations under noise conditions, such as the clipping of the low frequency masking vibrations.

In designing ships of large displacement, architectural acoustic problems arise which are closely related in their method of solution to the foregoing. A task of this type is that of ensuring optimal reverberation time and ensuring regularity of audibility in music halls, lecture halls and motion picture theaters. One of the methods of solution of this problem is acoustic treatment of the internal surfaces of the room, which simultaneously contributes to reducing noise in the room.

The main methods of controlling noise are the reduction of noise and vibration at the source, isolation and absorption of airborne noise and sonic vibrations (Figure 64). Considerable effect also may be obtained through rational location of ship compartments relative to each other, and the placement of noisy equipment in these compartments, taking into account the acoustic requirements in selection of the types and models of equipment.

Mention may be made of another group of methods which have an effect on the processes of origination and propagation of sound and vibration, and the processes of sound radiation. The measures deriving from these methods include prevention or elimination of resonant vibrations of structures and the impacting together of metallic parts, a reduction of the area of sound radiating surfaces, etc.

During recent years an active method for the deadening of low frequency tonal sounds has been applied in aviation and industrial construction in several countries, although on a limited scale. Sounds

may be weakened somewhat in individual parts of a room by this method, such as at the location of any permanently stationed machine operator.

The specific measures connected with the methods of isolation and absorption of sound may be divided into general, local and personal (with respect to man) measures.

The first group of measures includes sound isolation of entire rooms and groups of rooms, providing their internal surfaces with sound absorbing structures.

The second group of measures includes sound-isolating housing for machines, and the installation of various types of acoustic cabins, screens, enclosures and sound absorbing panels. Here we may include acoustic processing of ventillation ducts, installation of special ventillation baffles, engine exhaust and air intake mufflers, and the introduction of the use of sound absorbing structures in service systems and intra-ship transport systems.

Installation of the above-described sound-absorbing measures, however, often leads to a change in the architecture and the acoustic regime of the room as a whole. In cases of this type these means are correctly categorized as general sound-protection measures.

Among the personal measures for sound protection, up to the present time ear plugs placed in the aural canals of operators located in noisy sites, and sound-protection helmets and ear muffs have been proposed.

The most effective means for isolation of the sonic vibration of structures are sound isolation vibration mounts, elastic clutches and sockets, elastic inserts and strips of material having low acoustic resistance. In individual cases vibration-retarding masses may be utilized for isolation of bending vibrations in comparatively thin plates at frequencies above 200 to 400 cycles.

Absorption of sonic vibration in a broad band of frequencies is accomplished with the aid of special vibration-absorbing coatings applied to foundations and the hull structure. Coatings of this type have found increasingly extensive application in automobiles, aircraft and ships.

Vibration dampers or anti-vibrators are used for absorption of intense low frequency vibrations. They are installed on individual machines, bearings of drive shafts and decks. In principle, anti-vibrators consisting of an elastically connected added mass may be proposed also for absorption and isolation of bending vibrations at sonic frequencies (dampers of structure-borne sound).

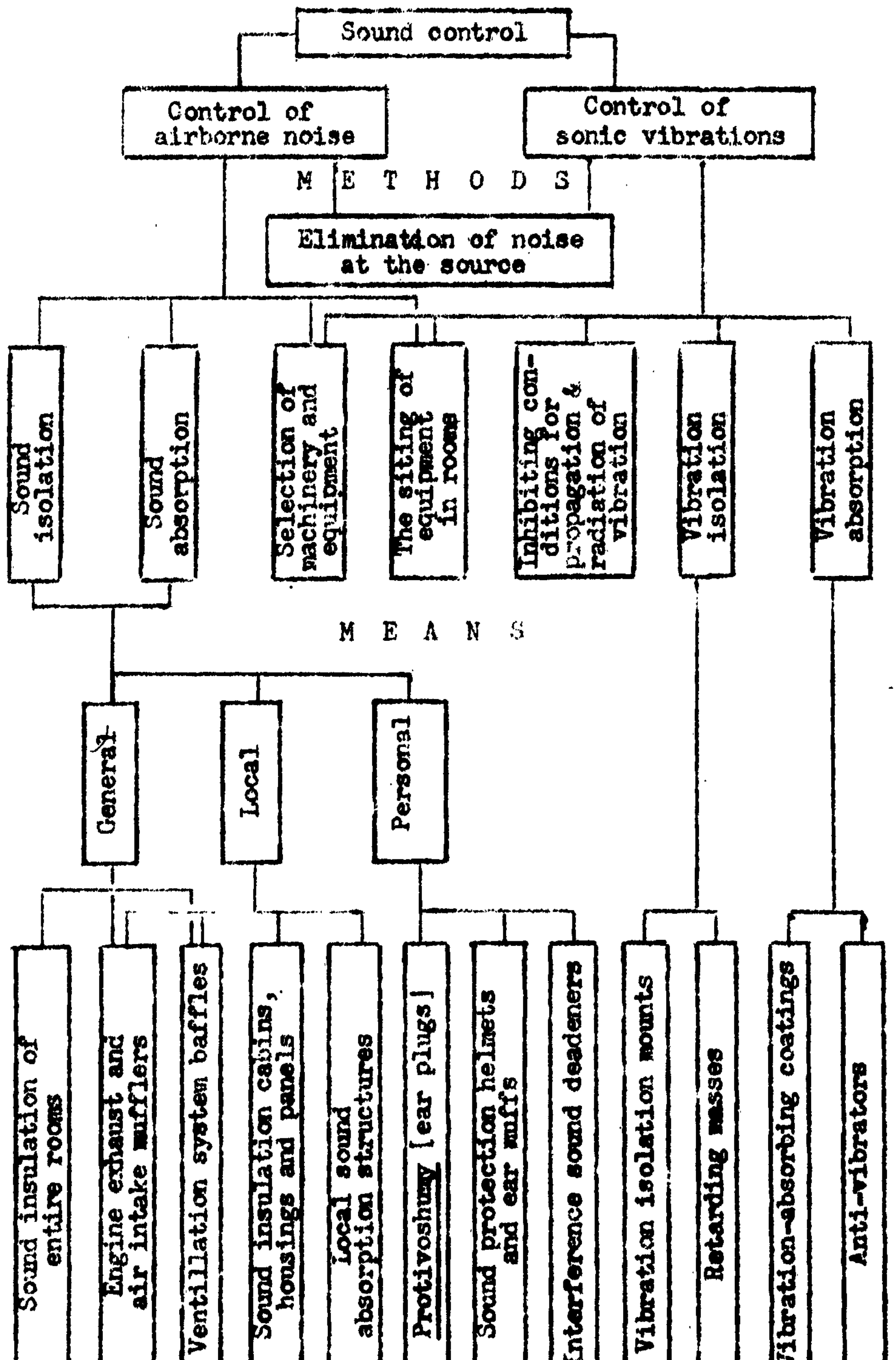


Figure 64. **Methods and means for controlling noise in ships.**

A list of the required sound protection measures is drawn up for each ship in the preliminary design stage. As a rule the desired acoustic effect is attained only with the simultaneous realization of a series of sound elimination measures. A considerable portion of the treatment for local sound protection may be installed in already completed ships during the period of their repair or modernization.

PART II

THE ISOLATION AND ABSORPTION OF NOISE AND SONIC VIBRATION IN SHIPS

CHAPTER 6

THE ISOLATION OF SOUND

23. Sound Isolation and Sound Absorption

The concepts of the "isolation" and "absorption" of sound sometimes are identified with each other in practice, although they differ in principle. A sound-isolating structure is intended to prevent the penetration of sound from one compartment into another, isolated, compartment. The absorption of sound in the isolating structure, itself, may be small, and its main effect is due to the reflection of sound from the structure (Figure 65, a).

The sound absorbing materials and structures serve to absorb sound both in the compartment of the source S (Figure 65, b), and in neighboring compartments (Figure 65, c). The absorption of sound is due to the transformation of vibratory energy into heat as a result of friction loss in the sound absorber. The loss due to friction is high in porous and loose fibrous materials, and because of this they are utilized in sound-absorption structures. Conversely, dense, solid materials are used for sound isolating structures.

The sound isolation method is more effective than the sound absorption method for reducing the sound level in rooms adjacent to the compartment containing a sound source. With the aid of sound isolating structures, the sound in neighboring rooms may be easily reduced by 30 to 40 db. Installation of a single sound absorber in a room, though it may have a very high degree of absorption, rarely results in a drop in sound level exceeding 6 to 8 db.

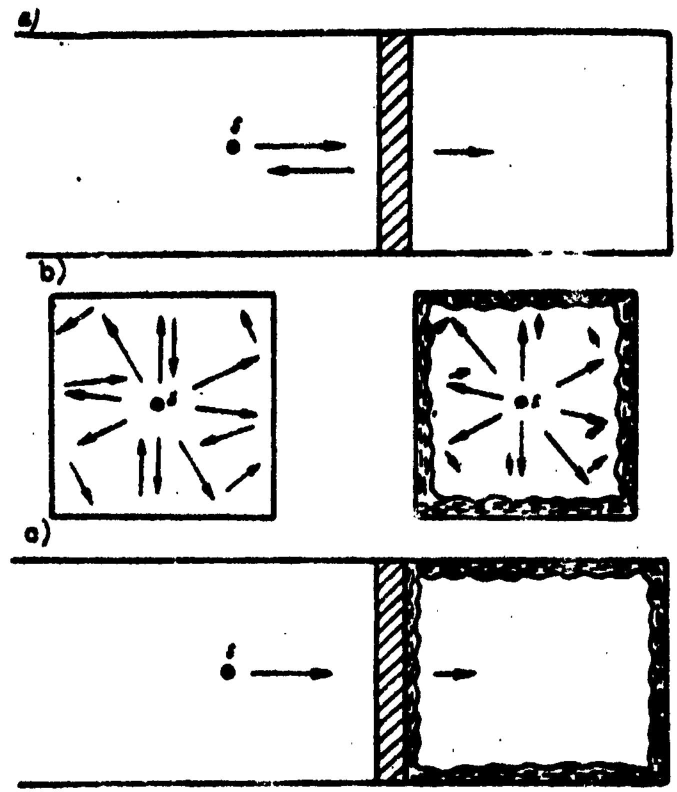


Figure 65. Defining the concepts of sound isolating and sound absorbing structures.

However, this significant effect of a sound isolating structure indicated is realized only because there is always a greater or lesser degree of sound absorption even in rooms not subjected to special acoustic treatment. If it were not for this absorption, the sound level under constant operation of the sound source would continue to rise until the useful effect of the sound isolating structure would drop to zero (cf. Paragraph 31). Thus effective sound protection requires simultaneous utilization of the methods of sound isolation and sound absorption, plus corresponding application of sound isolating and sound absorbing structures.

24. Sound Isolation of Main Bulkheads, Walls and Doors

Because the effect of sound isolation is based on its reflection, the isolation of sound in air, i.e. in a medium with low acoustic resistance, requires the use of barriers composed of materials with high acoustic resistance. Materials of this type may include metals, wood and solid plastic compositions (cf. Table 1).

The coefficient of transmission of sound impinging normally upon a boundary between two media or structures with different wave resistances or impedances is equal to (formula 85):

$$\alpha = 1 - \left|\frac{z_1 - z_2}{z_1 + z_2}\right|^2 .$$

In the case of a massive sound isolating bulkhead situated in air, its impedance z_1 is equal to the wave resistance of air $z_1 = \rho c$, and the impedance z_2 includes both the inertial resistance of the bulkhead per unit area (formula 29) and the wave resistance of the medium behind the wall, i.e.

$$z_2 = j\omega m + \rho c;$$

whence

$$\alpha = 1 - \left|\frac{-j\omega m}{2\rho c + j\omega m}\right|^2 .$$

Taking the modulus of the second member and performing simple transformations, we obtain:

$$\alpha = \frac{1}{1 + \left(\frac{\omega m}{2\rho c}\right)^2} \qquad (100)$$

The sound isolation of the bulkhead, expressed in decibels, is a value which is the reciprocal of α:

$$ZI_n = 10 \log \frac{1}{\alpha} = 10 \log \left[1 + \left(\frac{\pi f m}{\rho c}\right)^2\right] \text{ db.} \qquad (101)$$

At an oblique angle of impingment of sound on a bulkhead, its sound isolation is equal to (at frequencies far removed from the band of resonance of coincidence of the bulkhead, cf. Paragraph 11):

$$ZI_\theta = 10 \log \left[1 + \left(\frac{\pi f m \cos\theta}{\rho c}\right)^2\right] \text{ db.} \qquad (102)$$

where θ is the angle between normal and the angle of impingment of the sound upon the bulkhead.

Thus it is apparent that the sound isolation of a partition is lower at oblique angles of incidence than at normal. Under realistic conditions the sound field acting upon a partition is diffuse, i.e. all angles of incidence are of equal probability. This reduces the sound isolation in comparison to sound isolation at normal angle of incidence by a definite value Δ, i.e.

$$ZI = 10 \log \left[1 + \left(\frac{\pi f m}{\rho c}\right)^2\right] - \Delta \text{ db}.$$

If at fairly high frequencies we take the value $\Delta = 5$ db, as suggested by several authors (L. Cremer), and also ignore the units under the sign of the logarithm in comparison with the second term, then it is true for fairly high frequencies that after substitution of $\rho c = 41$ and performing the numerical transformations, we obtain

$$ZI = 20 \log (fG) - 46 \text{ db}, \tag{103}$$

where G is the weight per 1 m^2 of the bulkhead, in kg.

One other circumstance still has not been taken into consideration, viz. the effect of securing the edges of the bulkhead. Securing the bulkhead by its perimeter prevents its movement as a piston, i.e. reduces the active mass, which, consequently, is equal to $m' = \chi m$ $(\chi < 1)$. For a plate fixed at the edge we may take the value $\chi = 0.2$ (105). Taking account of the fact that $20 \log\left(\frac{1}{0.2}\right) = 14$ db, we finally find that:

$$ZI = 20 \log (fG) - 60 \text{ db}. \tag{104}$$

This formula is sufficiently well substantiated by practice, and because of this it is highly recommended.

Therefore the sound isolation of any ordinary barrier is proportionate to the logarithm of its mass or weight. Because of this, the above-described relationship of sound isolation is called the "law of mass."

Sound isolation also increases with frequency. On the basis of formulas (103) and (104) each two-fold increase in frequency would result in an increase of 6 db in the sound isolation. Actually, the increase is somewhat less. Here we encounter the limiting dimensions of real bulkheads, and other factors. At high frequencies of sound, the law of mass becomes invalid due to the resonance of coincidence, mentioned in Paragraph 11. This resonance arises when the length of

a bending wave in the bulkhead becomes equal to the length of the sound wave in air. At a frequency corresponding to the resonance of coincidence, and also at neighboring frequencies, the partition begins to conduct sound more strongly, i.e. its sound isolation drops. Figure 66 shows the experimental curve of sound isolation of a bulkhead of laminated plastic. At the frequency of resonance of coincidence, called the critical frequency f_{cr}, a sharp fall-off occurs in the curve of sound isolation, which up to this point had approximated the law of mass. The depth of this dip is greater with lower values of internal friction in the partition.

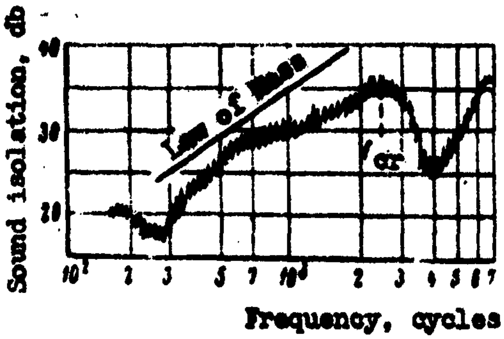


Figure 66. Experimental frequency curve of sound isolation of an ordinary, acoustically uniform partition.

A drop in sound isolation contrary to the law of mass also is observed at low frequencies as a result of resonant vibrations of the bulkhead as a membrane.

Figure 67 shows how the critical frequency varies with changes in the thickness of walls made of steel, and pine and beech boards. From the figure it may be seen that for a steel wall 3 to 6 mm thick and a wooden (pine) wall 2 to 5 cm thick, the critical frequency varies within a range of 5 - 2 kc, i.e. within the range of greatest sensitivity of the ear. With thinner walls the values f_{cr} increase correspondingly, and the resonance of coincidence is expressed to a lesser degree, because at high frequencies the force of internal friction in any given material is more strongly manifested. For walls of materials not indicated in Figure 67, the values of the critical frequency may be found from formula (76), using the modulus of elasticity and density of the material.

The multiplicity of factors determining the frequency function of sound isolation causes us to turn to the average values of sound isolation within a definite range of frequencies. In structural work, this range is taken as a 5-octave band of 100 to 3,200 cycles.

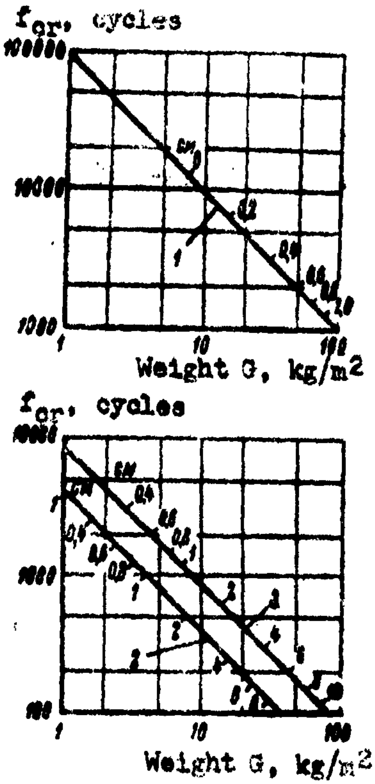


Figure 67. Critical frequencies of ordinary partitions of various materials.

1 - Steel partition; 2 - Partition of pine boards; 3 - Partition of beech boards.

The weight per m^2 of the barrier is plotted on the abscissa; the thickness of the barriers is indicated on the lines.

In many countries a series of semiempirical formulas has been suggested for determining the average sound isolation. The most precise is considered to be that proposed by A. K. Timofeyev (101, 30):

$$ZI_{av} = 13.5 \log G + 13 \text{ db.} \qquad (105)$$

The formula holds for ordinary, uniform walls of weight up to 200 kg/m^2. Other expressions of sound isolation have been suggested for heavier walls. However, because the cases of utilization of such heavy walls on ships are extremely rare, we do not include these expressions here.

Acoustically uniform structures include not only structures of any given single material, but also structures consisting of layers of various materials bound closely together (without an air layer).

In Figure 68 the broken line represents a graph of formula (105). Experimental points also are plotted according to data obtained by various investigators for the sound isolation of plywood panels of two thicknesses and a steel sheet 2 to 3 mm thick. The names of heavier structures are not indicated, although their points are plotted on the graph.

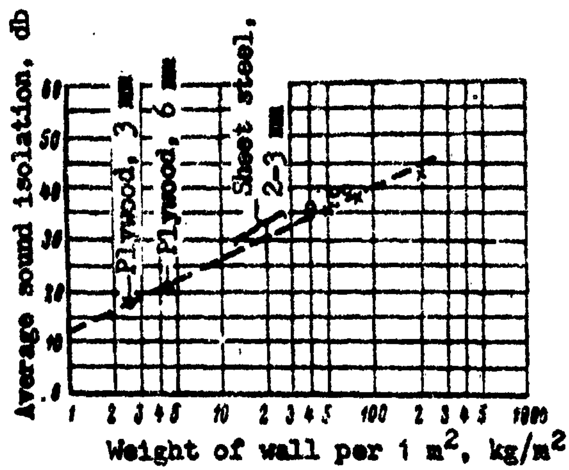


Figure 68. The average sound isolation in a frequency band as a function of weight for stiff, uniform walls.

Table 8 presents the average values of the sound isolation of various materials and structures in a frequency range of 100 to 3,200 cycles, obtained by experimental means by several authors. The table substantiates the foregoing comments on the high sound isolation properties of solid, non-porous materials and on the unsuitability of soft, porous materials for sound isolation purposes. Thus a felt slab 50 mm thick, having higher linear weight than, for example, sheet steel 0.7 mm thick, has less sound isolation value than the sheet metal. This is the result of the presence in the felt of cavities which form channels for the sound.

Table 9 (129) depicts the sound isolation of windows and doors, as used in construction work.

Table 8

Sound Isolation of Several Materials and Structures

Name of Material or Structure	Weight, kg/m^2	Average Sound Insulation db
Walls, Stiff Materials		
Double partition of 3-mm plywood, with 25 mm interspace filled with rock wool	8	26
Same, with 50-mm interspace	12	29
Same, with 65-mm interspace	14	34
Plywood, 3.2 mm, 3-layer	2.2 - 2.5	17 - 19
Same, 6.4 mm	4.5	21
Sheet steel, 0.7 mm	5.6	25
Same, 2 mm	15.7	33
Duralumin, 0.5 mm	1.8	15
Glass, 3 - 4 mm	8 - 10	28
Same, 6 mm	16	31
Fiberglass (glass mat Mark T, with PN-1 resin), 5 mm	--	23
Same, 15 mm	--	26
Soft Materials		
Wool cloth, 2 mm	0.05	5 - 6
Canvas	3.4 - 6.8	4 - 8
Hairy felt, 15 mm	2.8	6
Same, two layers	5.6	9
Same, four layers	11.3	17
Cardboard, 5 mm	3	16
Same, 20 mm	12	20
Cork slab, 50 mm	30	20

Table 9

Sound Isolation of Windows and Doors

Type of Window	Thickness of Glass, mm	Average Sound Isolation db
Window with single pane	3 - 4	22 ± 2
Window with double pane	2 x (3-4)	26 ± 2
Window with double thickness of glass, with gasket	5 - 7	32 ± 2
Type of Door	**Weight per 1 m^2, kg**	**Average Sound Isolation, db**
Light single wooden door without gasket, ordinary latch	6	24 ± 3
Single wooden door without gasket, ordinary latch	15	27 ± 2
Double door, consisting of two doors of type 2, with 40 cm between latches	30 - 35	33 ± 3
Door of type 3, with felt gasket	35	36 ± 3

Similar data were obtained according to measurements conducted under shipboard conditions by the present author and V. M. Kriger (Table 10). The table indicates the sound isolation values of ship doors, windows and portholes. It is apparent from the table that a door of thin steel, not watertight, has a relatively low sound isolation value, approximately 25 db. The sound isolation of watertight doors and various shipboard light-admitting structures exceeds 30 db.

Turning again to a uniform sound isolating partition, several factors in addition to the weight of the partition which determine the frequency characteristic of the curve of sound isolation and its average value must be mentioned. Factors of this type include the presence of grooves (not penetrating through the partition) and ribbing, the means by which the partition is secured at the periphery (degree of fixity), and the magnitude of mechanical losses in the partition.

Table 10

Sound Isolation of Several Designs of Shipboard Doors, Windows, Ports and Hatches

Name	Brief Description	Average Sound Isolation, db
Ship door, nonwatertight	Dimensions 890 x 1700 mm. Consists of sheet steel 1.5 mm thick, with two pressed rectangular depressions for structural strength. The door is protected on the internal side by 5 mm plywood. Between the steel and the plywood is a 50 mm layer of Alfol aluminum foil for insulation. Insulation is absent at the site of the latch and knob.	25
Ship door, watertight	Dimensions 680 x 1600 mm. Consists of sheet steel 3 mm thick with reinforcing ribs and 5 mm felt insulation. The door is protected on the internal side by a 5 mm plywood panel. There are 5 sliding bolt latches, and a rubber gasket at periphery.	32
Opening port	Dimensions 450 x 650 mm. Double glass panes 5 mm thick, with 20 mm interspace. The frame [illegible] which the glass is fixed has reinforcing ribbing and a rubber gasket at the periphery.	37
Fixed port	Diameter 200 mm. The port has a single glass pane, 5 mm thick. The mounting is provided with reinforcing ribbing and rubber gasket.	34
Cover of light-admitting hatch of engine room	Consists of 3 mm sheet steel, covered from within by a layer of shredded felt 5 mm thick. Six glass ports 200 mm in diameter are installed in frames in the sheet, with 5 mm-thick glass. The hatch has reinforcing ribbing and rubber gasket at periphery.	31

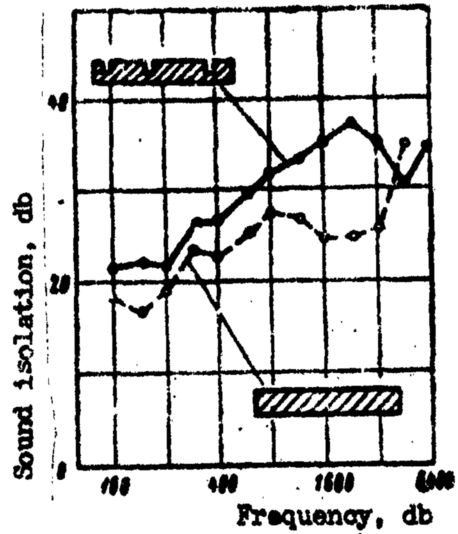


Figure 69. Sound isolation of plywood panels without, and with grooves.

Figure 69 presents the sound isolation curves (124) of a solid plywood sheet 15 mm thick, and a sheet of the same thickness containing grooves (not slits). Increasing the sound isolation of the sheet by inclusion of grooves is based on reduction of the elasticity of the sheet, i.e. increasing its critical frequency (cf. formula 76). The partition begins to behave as a series of individual masses, and its sound isolation approximates the law of mass.

Although reduction of the stiffness of a partition often increases sound isolation and it may be expected that, when increasing its stiffness without changing the weight (such as through the addition of reinforcing ribs), it is found that on the contrary, this deteriorates its sound isolation quality. This has been substantiated by experience (Yu. I. Shneyder). Welding of light ribs onto a steel partition results in a reduction of 3 to 5 db in its sound isolation at frequencies above 1,000 cycles (Figure 70). Welding of heavy ribs onto the same partition deteriorates its sound isolation at all frequencies above 200 cycles by a value equal to 9 db (curve 3). One of the causes for the drop in sound isolation is the reduction of the critical frequency of the partition. Thus the welding of steel ribs onto a steel partition, which at first glance appears to be a very useful measure for changing the acoustic quality of the partition, cannot be recommended for practical use.

The sound insulation of partitions is influenced by the manner in which they are secured at the edges. As indicated by the investigations of M. Heckl (137), in partitions rigidly fixed at the periphery sound isolation decreases at frequencies above the critical level as a result of transmission of sound energy through the point of the fixation. Because of this, the use of a rubber seal at the periphery of a steel partition (cf. Figure 99) increases the average sound insulation in the standard frequency range by 2 to 5 db in comparison to a partition rigidly secured at the edge.

The magnitude of mechanical loss in the partition has an effect mainly upon low sound frequencies, at which membrane vibration of the

partition occurs. In the very simple case of a square homogeneous plate with free edges (this case approximates that of elastic securing of a partition at the periphery), the frequency of natural vibrations may be found according to formula (6):

$$f_{н} = \frac{\zeta}{a^2}\sqrt{\frac{D}{m}} \text{ cycles,} \tag{106}$$

where a is the side of the square;

$m = \frac{\gamma h}{g}$ is the mass of the plate per unit area;

h is the thickness of the plate;

γ is the specific weight of the material;

g is the acceleration due to gravity;

$D = \frac{Eh^3}{12(1-\mu^2)}$ is the flexural rigidity of the plate;

E and μ are the Young's modulus and Poisson's coefficient, respectively, of the material of the plate.

The coefficient ζ for the three lowest tones of vibration of the place has the following values: $\zeta_1 = 14.1$; $\zeta_2 = 20.6$; $\zeta_3 = 23.9$.

In the case of plates fixed along their periphery, the values of ζ are somewhat higher.

Calculations according to formula (106) give fairly low natural frequency values for actual bulkheads, in the range of tens or several tens of cycles. At these frequencies the internal friction in any given structural material is weakly manifested, and because of this the corresponding frequency components in the noise spectrum of a machine are poorly isolated by the bulkhead. Intense vibration of the bulkhead is observed when the bulkhead is rigidly fixed to a source of low frequency exciting forces.

In this case double-layer bulkheads may be of help, in which surface friction is utilized during flexural vibrations. An example of a bulkhead of this type is shown in Figure 71-a. The plates of the bulkhead are joined together with spot welds, welded in perforation, or slots in one of the plates. Spot welds must not be used too much because this would result in the possibility of co-movement of the plates of the bulkhead in flexural vibration.

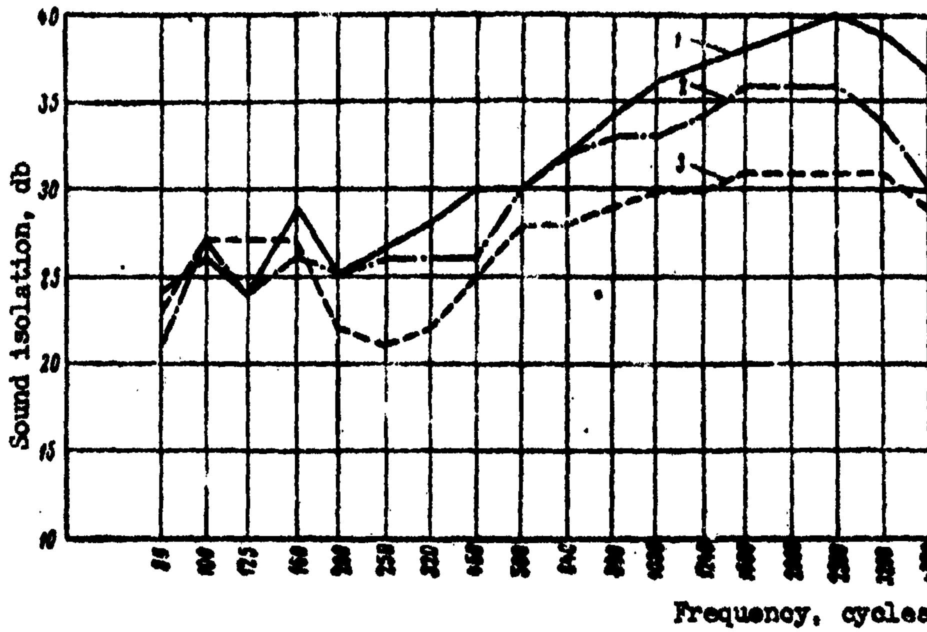


Figure 70. Influence of stiffeners on the sound isolation of a steel bulkhead.

1 - Steel bulkhead 3 mm thick, without stiffeners; 2 - Same, with the addition of 7 welded stiffeners consisting of bulb bar No. 6; 3 - Same, with addition of 7 welded bulb bars No. 12.

If the plates have an uneven surface and are not secured adequately to each other, surface friction will be low. In this case the damping action of the bulkhead may be increased somewhat by injecting a damping composition between the plates, such as red lead, pitch, etc. For this purpose holes are drilled in one of the plates at various places and nuts are attached for the insertion of the injection nozzle (these nuts serve to secure a sound absorber or a decorative covering). Effective lessening of vibration of the bulkhead is attained when the damping composition occupies a definite part of the area of the partition, approximately one-third of its total area.

During recent years the practice of using bulkheads utilizing the internal friction of elasto-viscous materials has been introduced in foreign shipbuilding. The partitions have a "sandwich" structure (Figure 71-b) and consist of two steel sheets with a solid plastic composition between them. It has been shown (144) that with very high

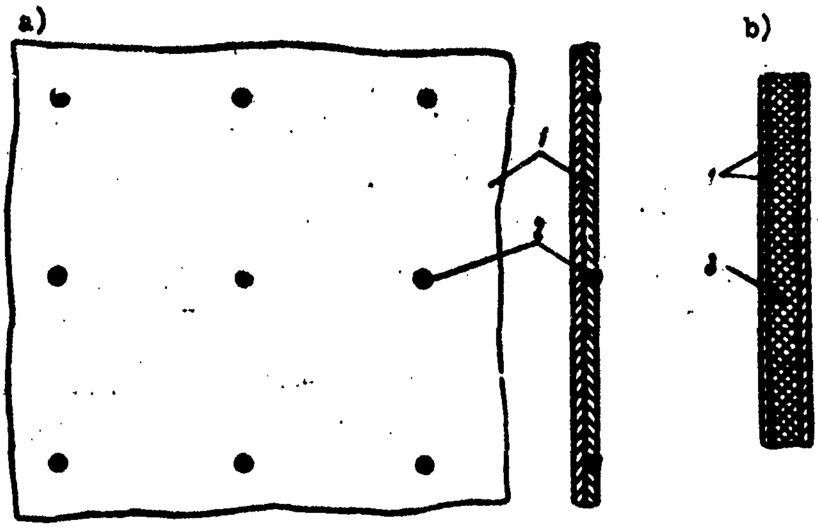


Figure 71. Double-layer ship's bulkhead (a), and a "sandwich" type laminated bulkhead (b).

1 - Steel sheets; 2 - Spot welds; 3 - Plastic composition layer.

loss values in the sandwiched layer a laminated partition of this type is able to undergo vibration in shear, but not flexural vibration, at high frequencies, with the result that the deterioration of sound isolation at resonance of coincidence practically disappears and sound isolation at high frequencies is close to the law of mass. The resonance of membrane vibration also is lessened.

The static rigidity of a ship bulkhead of this type may be fairly high. Thus it is emphasized that combinations of two 1-mm steel sheets with a sandwiched layer of solid plastic composition 4 mm thick has the same static rigidity as a solid steel partition 5 mm thick (145). Partitions of this type have been used successfully on the Netherlands ship "Queen Wilhelmina," in which noise in the passenger saloon has been considerably reduced (165). There is no doubt that laminated partitions will find application in those rooms in which especially good sound isolation is needed.

To the problem of isolation of airborne noise by bulkheads and decks is added the problem of the reduction, by them, of noise arising from impact excitation, such as in walking to and fro on the deck. Table 11 (18) presents the average values of the reduction of noise by ordinary deck coverings of various materials (relative to wooden planking). The reduction of noise results from the sound isolating and

vibration damping action of the materials and structure of the covering. Fairly large values of reduction of noise are obtained with the use of elastic matting and materials with high internal friction, such as asbestos-cement slabs and kordin. The latter consists of slabs of spun cotton thread waste and shredded rubber obtained from worn-out automobile tire casings. The interesting work of V. I. Zaborov (38) substantiates the conclusion relative to damping impact noise with isolating coatings. The damping of ship hull structures by means of materials with high internal friction is discussed in greater detail in Chapter 14.

Table 11

Influence of Deck Coverings on the Level of Impact Noise Underneath the Deck

Structure and Material of the Covering	Reduction of the level of impact noise underneath the deck, relative to the level for the case of a wooden plank covering, db
Parquette floor on asphalt 20 mm thick	0
Linoleum, 2.5 mm	2
Rubber sheeting, 5 mm	3
Asbestos-cement slabs ($\gamma = 350$ kg/m^3), 30 mm thick	5
Slabs of mineral felt ($\gamma = 300$ to 350 kg/m^3), 30 mm thick	4
Porous wood-fiber slabs ($\gamma = 200$ to 250 kg/m^3), 25 mm thick	2
Kordin slab covering ($\gamma = 320$ to 350 kg/m^3), 35 mm thick	6
Springy elastic coverings	10 - 12

25. Sound Isolation of Double Partitions

The sound isolation of a boundary structure may be increased substantially without changing its weight through use of the double partition design with intermediate air space. The useful acoustic effect of the air space appears at average and high sound frequencies (Figure 72). This results from the multiple reflection and accompanying absorption of sound in the interstice.

At low sound frequencies the sound isolation of a double partition usually is slightly lower than that of a single partition. At

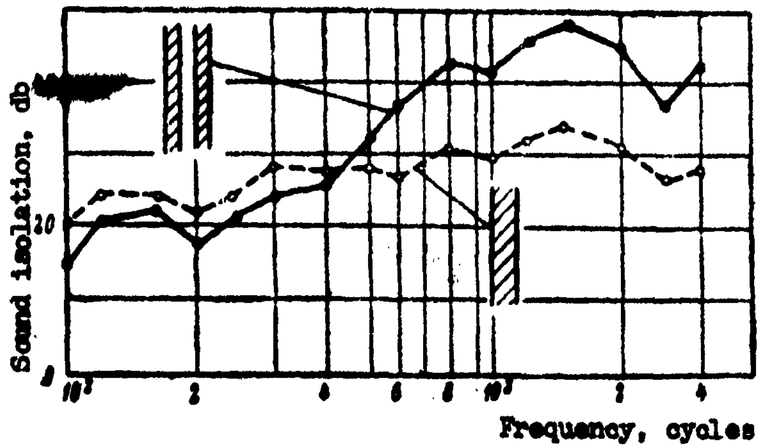


Figure 72. Sound isolation of a single and double partition.

these frequencies resonance of the partition is observed, consisting of a system of two masses m_1 and m_2, joined by the elasticity c of the air volume between the partitions. The natural frequency of a system of this kind is:

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{C(m_1+m_2)}{m_1 m_2}} \text{ cycles.}$$

The elastic force during compression of the air volume V is equal to (60):

$$F = \rho c^2 \frac{s^2}{V}\xi,$$

where s is the area of the basic volume;

ξ is the deformation.

The elasticity, i.e. force per unit deformation, is equal to (for $s = 1$ cm^2):

$$C = \frac{F}{\xi} = \frac{\rho c^2}{h};$$

here h is the height of the air volume, in the given case the air layer between the walls of the double partition (Figure 73).

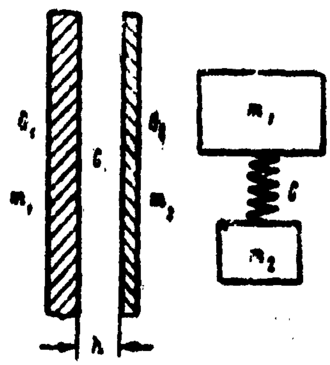


Figure 73. Determination of the fundamental resonance frequency of a double partition.

The mass m per 1 cm^2 of a partition (g) is related to its weight G (kg) per 1 m^2, by the expression $m = 0.1\,G$.

Substituting the expressions for c and m in formula (106), we obtain:

$$f_0 = 600\sqrt{\frac{G_1 + G_2}{G_1 G_2 h}} \text{ cycles.} \tag{107}$$

For equal weight of the walls,

$$f_0 = \frac{850}{\sqrt{Gh}} \text{ cycles} \tag{108}$$

(G is expressed in kg/m^2, and h in centimeters).

The expressions (107) and (108) for f_0 are based upon the assumption of piston movement of the walls. When the walls undergo flexural vibration, their effective mass is less. Because of this, the original values of f_0 may be somewhat greater than those found according to formulas (107) and (108).

In addition to fundamental resonance of a double partition, resonance of the air layer also may occur. This resonance appears at frequencies at which a whole number of sonic half-waves can be accommodated across the width of the air layer. The frequency of the first resonance is equal to:

$$f_{1fr} = \frac{c}{2h} \text{ cycles.} \tag{109}$$

The greater is the width of the air layer h, the lower is the frequency at which wave resonance of the layer begins to appear, i.e. the greater is the number of resonances which appear in the given band of frequency. On this basis several German pre-War investigators (Kammerer and Durchammer) stated that there is an optimum magnitude of air layer between walls. Later investigations, however, did not substantiate this theory. Within the limits of practical dimensions for double partitions, the average sound isolation in a band increases with the increase in width of the layer between them (Figure 74).

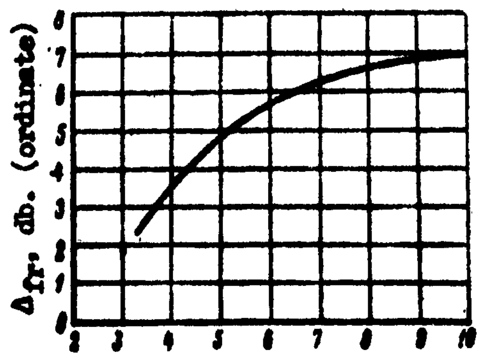


Figure 74. Sound isolating ability of the air layer of a double partition.

The average sound isolation in the frequency range of 100 to 3,000 cycles of a double wall with an air layer may be determined with a sufficient degree of precision from the expression:

$$ZI_{av} = 13.5 \log (G_1 + G_2) + 13 + \Delta_{fr} \text{ db}, \qquad (110)$$

where $G_1 + G_2$ are the weights per m^2 of the double partition;

Δ_{fr} is the sound isolation of the air space between the walls of the partition, determined from Figure 74.

Comparison of formulas (105) and (110) indicates that the sound isolation of a double wall is greater than the sound isolation of a single wall of weight $G_1 + G_2$ per m^2 by the amount Δ_{fr}. We may note the great role of double sound isolating partitions in ships, due to the great relative importance of structure-borne sound. A single partition isolates air-borne sound, but apparently is in essence a radiator of sound due to the vibrations of the boundary structures (Figure 75-a). A double partition isolates both air-borne sound impinging on it, and sound radiated due to the vibration of the first part of the partition (Figure 75-b). From this it follows that it is

especially advantageous to utilize double bulkheads at sites of intense sonic vibration, i.e. for example, in compartments immediately adjacent to the engine room. In this case supplementary masses are attached to the second bulkhead to increase its sound isolation (164) (Figure 75-b), i.e. in essence, that bulkhead is changed into a kind of sound isolating wall with grooves (Figure 69).

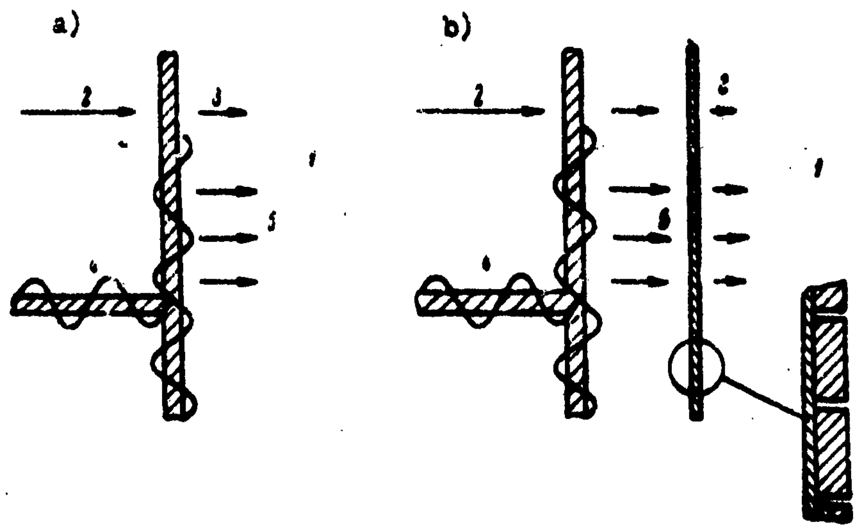


Figure 75. Acoustic effect of a single (a) and double (b) partition under the simultaneous effect of airborne noise and sonic vibrations.

1 - Isolated compartment; 2 - Airborne sound impinging on the isolating structure; 3 - Sound penetrating into isolated room: 4 - Sonic vibration; 5 - Airborne sound generated by sonic vibrations.

Example 24. In planning a shipboard sound isolating bulkhead, it is given that the most intense component of noise which must be isolated is in the frequency range of 125 to 150 cycles. The problem is to determine which double bulkhead is preferable, a double bulkhead of 3-mm sheet steel (with 5 cm air layer), or a composite double bulkhead consisting of a 3-mm steel sheet and 6.4-mm plywood sheet. The average sound isolation of the selected bulkhead also must be determined.

Solution. The weight per m^2 of the 3-mm sheet steel, $G_1 = 23.6$ kg, and the weight per m^2 of the 6.4-mm plywood is $G_2 = 5$ kg. The fundamental frequency of the double bulkhead made of two steel sheets (formula 108) is:

$$f_0 = \frac{850}{\sqrt{23,6 \cdot 5}} = 78 \text{ cycles.}$$

The fundamental frequency of the composite bulkhead consisting of a steel sheet and plywood (formula 107) is:

$$f_0 = 600\sqrt{\frac{23,6+5}{23,6 \cdot 5 \cdot 5}} = 132 \text{ cycles.}$$

The fundamental resonance of the composite bulkhead lies within the range of the most intense components of the spectrum of the noise which is to be isolated, and because of this selection of the sheet steel bulkhead is to be recommended. The average sound isolation of this bulkhead (formula 110 and Figure 74) is:

$$ZI_{av} = 13.5 \log (2 \cdot 23.6) + 13 + 5 \cong 40.5 \text{ db.}$$

26. The Effect of Slits and Openings on Sound Isolation

Slits and openings in partitioning structures have considerable effect upon their sound insulation, especially at high sonic frequencies. Figure 76 is a schematic representation of the propagation of a sound wave through openings, with various ratios between the wave length λ and the diameter of the opening.

With a large transverse dimension of the opening a in comparison to the wave length, the front of the waves passing through the opening will be flat (Figure 76-a), and thus all the sound impinging on the opening passes through it. The situation is different with a high ratio $\frac{\lambda}{a}$. The wave passing through the opening is spherical (Figure 76-b), and because the impedance of a plane wave falling upon the opening, and of a spherical wave passing through the opening differ in value, considerable reflection of the wave is unavoidable. Thus the sound transmissivity of small openings is not very great up to very high sound frequencies (55). From this it follows also that 100 small openings, for example, distributed on the surface of any given partition, affect its sound isolation less adversely than a single opening with area equal to the total combined areas of the small openings. This is substantiated by the sound isolation curves (Figures 77 and 78) plotted according to the measurement data of Yu. I. Shneyder. Figure 77 compares the curves of sound isolation of partitions without openings, and of partitions with openings distributed irregularly and compactly. With irregular distribution of the openings no loss in noise isolation is noted up to a frequency of 700 or 800 cycles. At

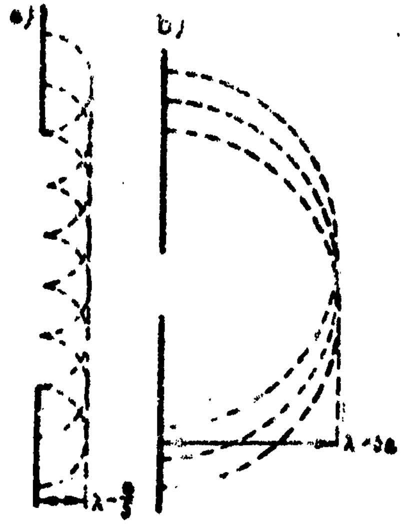


Figure 76. Determining the sound transmissivity of openings of various diameters

frequencies of 2 or 3 kc the reduction in sound isolation is equal to 4 or 5 db. Concentration of the openings in one area leads to a drop in sound isolation of from 2 or 3 to 7 or 8 db in the entire range of frequencies.

Figure 78 enables comparison of the loss of the sound isolation property of partitions caused by openings or slits of identical area. A plane wave is transformed into a spherical wave behind the circular hole, and is transformed into a cylindrical wave behind the slit. The impedance of a cylindrical wave is closer to the value of the impedance of a plane wave than is the impedance of a spherical wave. Because of this the reflection of sound is less, and the transmission of sound is greater in the case of the slit than in the case of the hole. Actually the loss of sound isolation resulting from the slit is greater at all frequencies than that resulting from a hole of identical area.

For evaluation of the loss of sound isolation ΔZI caused by a hole, we apply the following formula:

$$\Delta ZI = 10 \log \left(1 + n\frac{S_0}{S}\, 10^{0.1 ZI}\right) \text{ db}, \tag{111}$$

where S_0 is the area of the hole;
S is the area of the partition;
ZI is its sound isolation.

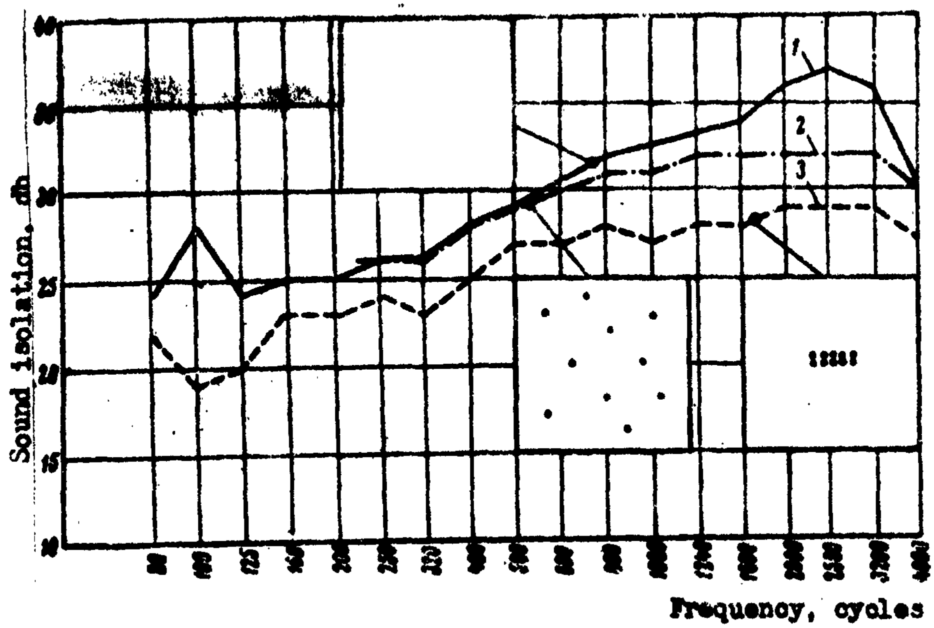


Figure 77. Effect of the distribution of openings in a partition upon its noise isolation.

1 - Partition 2 x 2 m, without openings; 2 - Same partition with 10 openings of 11 mm diameter, distributed irregularly over the entire area of the partition; 3 - Same, with openings concentrated at the center of the partition.

The value of the coefficient n depends upon the frequency.

Frequency, cycles	800	1200	1800
Coefficient n	12	6	2.5

The formula is derived for openings with diameter up to 12 or 15 mm, although it holds approximately for openings of greater diameter, also. Although it yields insufficiently precise frequency functions of the loss of sound isolation, it has the interesting and practically important property of revealing that a greater sound isolation value of a partition corresponds to a greater value of ΔZI. This peculiarity is apparent from the following example.

<u>Example 25</u>. Two bulkheads, each with dimensions of 2 x 2 m, have sound isolation values of 40 and 25 db, respectively, at a frequency of 1,200 cycles. The problem is to determine the drop in sound isolation in each of the bulkheads when holes 15 mm in diameter are bored in them.

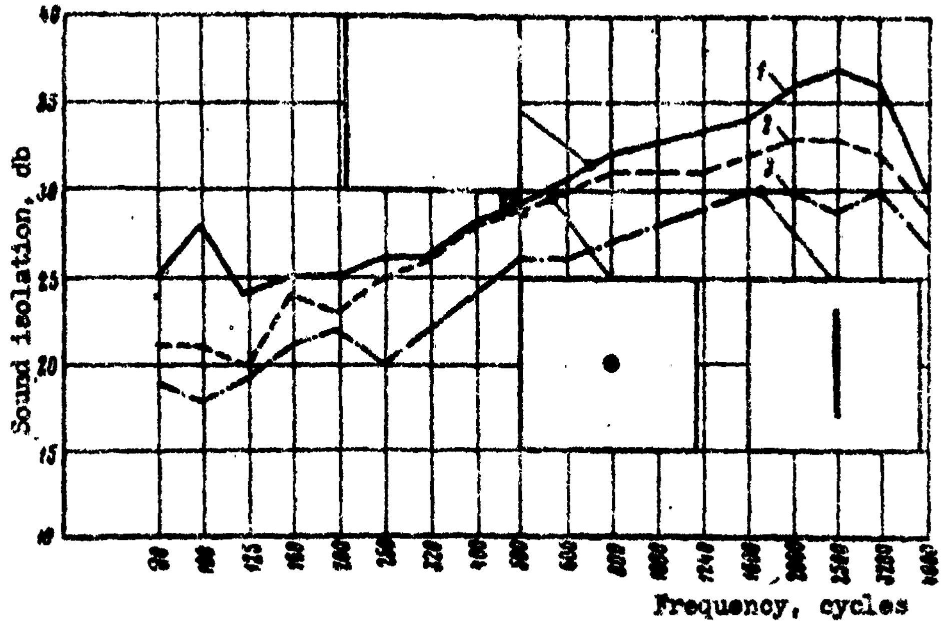


Figure 78. Effect of circular holes and slits of identical area upon the sound isolation of a partition.

1 - Partition without holes or slits; 2 - Partition with one hole 25 mm in diameter; 3 - Partition with slit 1 x 500 mm.

Solution. The loss of sound isolation of the first bulkhead is:

$$\Delta ZI_1 = 10 \log \left(1 + 6\,\frac{\frac{\pi}{4}(1,5)^2}{4\cdot 10^4}\,10^{0,1\cdot 40}\right) = 5,6 \text{ db};$$

and the loss for the second bulkhead is:

$$\Delta ZI_2 = 10 \log \left(1 + 6\,\frac{1,8}{4\cdot 10^4}\,10^{0,1\cdot 26}\right) < 0,5 \text{ db}.$$

The presence of wide slits or large holes in sound isolating structures leads to very significant losses in sound isolation. From Figure 79 it may be seen how the average sound isolation of a door made of boards 25 mm thick changes with changes in the width of slits between the door and the door jamb or floor. The data were obtained by

experimental means (30). It is apparent that with poor alignment of the door the decrease in its sound isolation attains 6 to 10 db.

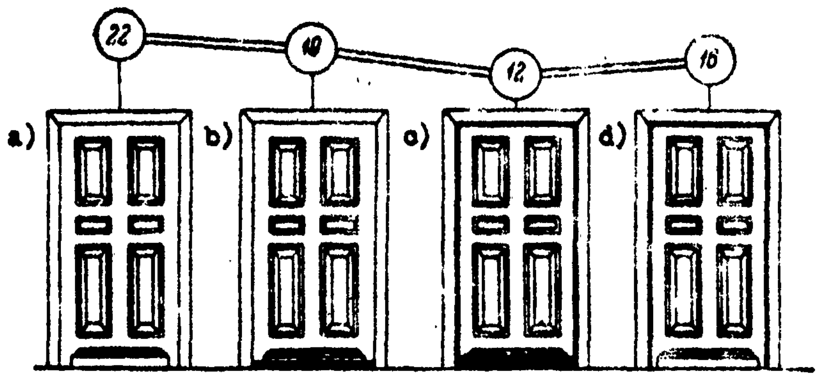


Figure 79. Effect of tightness of closure of a door and of the size of slits between the floor and the door upon the average sound isolation of the door (sound isolation values are indicated in db in the circles); a) - A well aligned door with slits less than 0.5 cm wide; b) - Same, with 1-, to 1.2-cm slits; c) - Door with poor seal at the periphery, and slit at bottom 1.5 to 1.8 cm wide; d) - Same, with slit at bottom absent.

Example 26. The problem is to compare, on the basis of the data of Figure 79, the sound transmissivity of the wooden door and that of the slits at its periphery.

Solution. We shall assume that a narrow slit (Figure 79-a) changes the sound transmissivity of the door very little, i.e. that the average sound isolation of the door itself is approximately 22 db. With expansion of the slit at the bottom and the appearance of slits at the periphery of the door (Figure 79-c) the sound isolation of the door decreases by 22 - 12 = 10 db, i.e. the amount of sound energy penetrating into the neighboring room increases 10-fold. Therefore, the sound transmissivity of the slits at the periphery in the given case surpasses the sound transmissivity of the door 10-fold.

If the closure of the door is tight but there is a wide slit at the bottom, it may be seen from Figure 79-b that the sound isolation is 3 db lower than the sound isolation of the door, itself, i.e. the sound energy penetrating through the wooden structure increases two-fold in the presence of a wide slit. We may draw the conclusion which is at the same time a good mnemonic rule and saying, that a chink under a door permits the passage of the same amount of sound as the door itself.

CHAPTER 7

SOUND ABSORPTION

27. Sound Absorption Properties of Materials for the Case of Normal Angle of Incidence

The sound absorption of materials and structures is evaluated by means of the coefficient of absorption α, which is the ratio of the sound energy absorbed by the material to the energy of the incident sound. Values of energy may be replaced by the corresponding values of the intensity of sound. Thus we have:

$$\alpha = \frac{J_{abs}}{J_{inc}} . \tag{112}$$

Expressed as the ratio between the sound pressures of the reflected and incident waves, the coefficient of absorption is equal to (cf. formulas 83 - 85):

$$\alpha = 1 - \left(\frac{p_{ref}}{p_{inc}}\right)^2 = 1 - \beta'^2 \tag{113}$$

where β' is the coefficient of reflection of sound pressure.

The coefficients of reflection and absorption of noise may be determined most simply in the case of normal (perpendicular) angle of incidence of sound with respect to the surface of the material. For measurements of this type, the apparatus diagrammed in Figure 80 may be used. The test sample of sound absorbing material or structure is secured at the end of a fairly long tube, in which standing sound waves are generated. By means of a microphone moving uniformly along the length of the tube the sound pressure at all points of the standing wave may be determined, especially the maximum and minimum values of the sound pressure. If a material completely reflecting sound is located at the end of the tube, the minimum pressure (at the node of the standing wave) will be equal to zero. In the case of material which absorbs sound, the pressure at the "node" will be other than zero (strictly speaking, the term "node" is inapplicable in such a case).

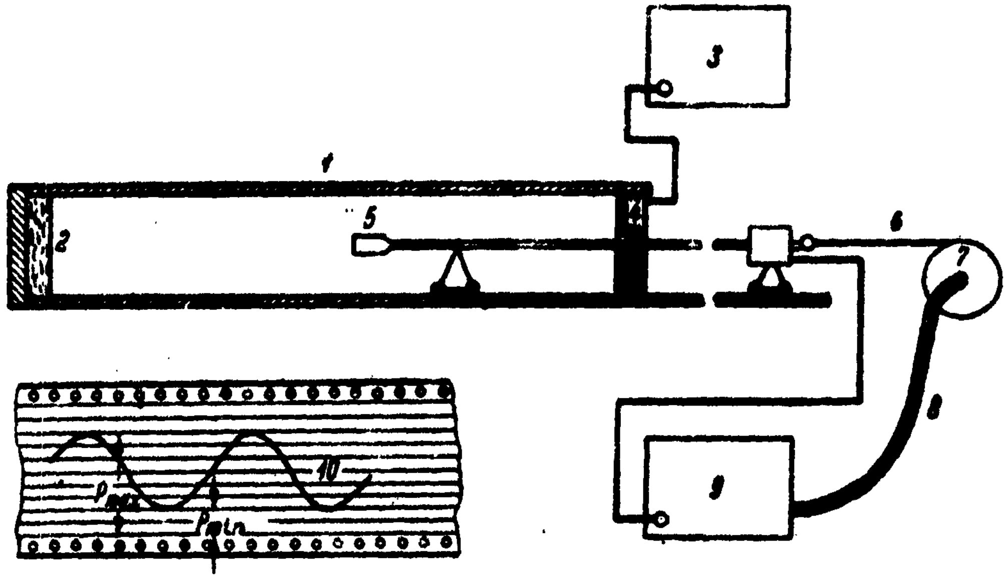


Figure 80. Measurement of the coefficient of sound absorption of a material in a tube.
1 - Measuring tube; 2 - Sample of sound absorbing material; 3 - Sound generator; 4 - Radiator; 5 - Point microphone; 6 - Thread wound on drum; 7, 8 - Flexible shaft; 9 - Pen recorder; 10 - Sample of recording of sound pressure on the pen recorder.

The lower is the value of the ratio of the maximum to minimum amplitude of sound pressure

$$n = \frac{p_{max}}{p_{min}}, \tag{114}$$

the greater is the coefficient of sound absorption of the sample of material at the end of the tube. This radio may be measured with the aid of a needle indicator attached to the output of the microphone amplifier, or may be determined directly according to the recording of a level recorder, which in the given case utilizes a linear (not logarithmic) potentiometer. The coefficient of sound absorption of the material is equal to:

$$\alpha = \frac{4}{2 + n + \frac{1}{n}}. \tag{115}$$

There are some instruments which indicate the value of α directly on a scale, entered with a known value of n.

The method described, referred to as the tube method, is suitable for the purpose of comparing the acoustic properties of small samples of sound absorbing materials, but other methods must be used for determining the true value of sound absorption of these materials in the sound field of a room.

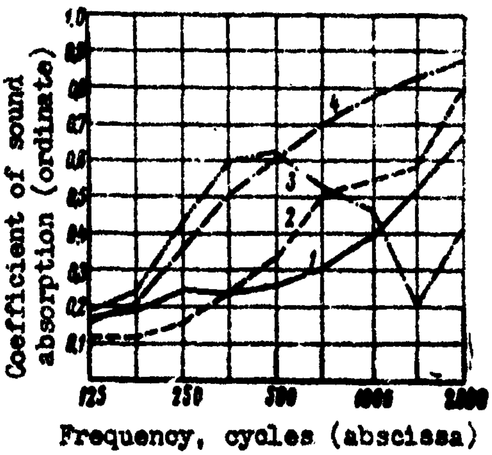


Fig. 81. Coefficients of sound absorption of several acoustic materials with normal angle of incidence of sound (thickness of material 40 mm). 1 - Asbopukhshnur [asbestos floss lacing] (fluffy); 2 - Glass felt; 3 - FS-7 material (with open pores); 4 - Mineral felt.

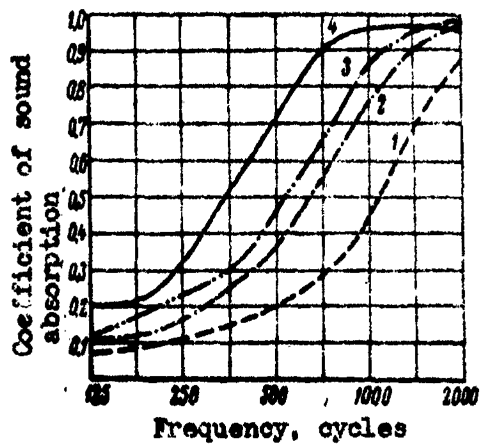


Fig. 82. Coefficients of sound absorption of caprone felt VT-4S as a function of the thickness δ of layer. 1 - δ = 20 mm; 2 - δ = 30 mm; 3 - δ = 40 mm; 4 - δ = 60 mm.

Figures 81 and 82 show the frequency curves of the coefficient of absorption obtained by the tube method (2) for several domestic materials frequently used in sound-protection structures. Mineral felt, caprone [nylon] fiber and FS-7 material (on a base of fiberglass) have fairly high sound absorption values.

With an increase in the thickness of the material, sound absorption begins to appear at lower frequencies (Figure 82). This is explained by the fact that for sound absorption, the length of path relative to the wave length, and not the absolute length of the path of sound in the material is of importance. With an increase in the

thickness l of the sound absorber, the frequency at which the ratio $\frac{l}{\lambda}$ is maintained is decreased.

For each material there is a certain limiting thickness l_{lim}, the exceeding of which is disadvantageous because it does not lead to a noticeable increase in absorption. This limiting thickness depends upon the degree of blow-through resistance of the material. The blow-through resistance is determined with a special apparatus, in which a constant current of air is blown through a slab of the material tested (21, 98). The limiting thickness of sound absorbing layers of a series of materials computed on the basis of their blow-through resistances are presented in Table 12 (42). As may be expected, the greatest value l_{lim} is exhibited by the porous fibrous materials such as cotton wadding and felt. Therefore, with sufficient thickness, absorbers may be made with these materials which are not only very effective, but which also have a very broad range of frequencies of absorption.

Table 12

Upper Limits of Useful Thickness of Several Sound Absorbing Materials

Material	l_{lim}, cm
Raw cotton, waddi.g	40 - 80
Loose wool fiber	18
Dense wool fiber	12
Mineral wadding	9
Test slab	7.5
Paper cardboard	1.8
Porous plaster	0.6

The frequency curves of sound absorption (Figures 81 and 82) relate to the case in which the absorbing material is applied to the surface of a solid partition (Figure 83-a). If the sound absorber is secured with a space between it and the partition, or as is sometimes said, "at an offset" (Figure 83-b), increased sound absorption may occur at lower frequencies. This is due to a rather large ratio $\frac{l}{\lambda}$ in comparison to the case of applying the material directly to the partition. In addition, due to the air space between the absorber and the partition, the phenomenon of resonance also begins to take a role. Reduction of the frequency of the lower limit of effective absorption enables a very efficient sound protection, with relatively low weight of sound absorber, in the range of frequencies most sensitive to the

sense of hearing (in Figure 83 this range is indicated by dotted lines).

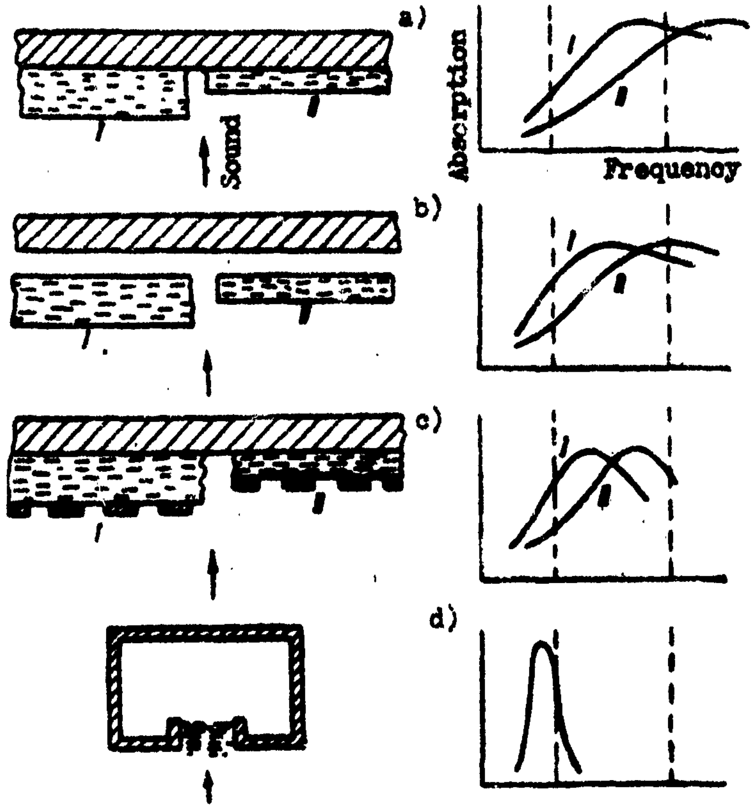


Figure 83. Methods of application of sound absorbing material, and their corresponding frequency absorption characteristics.

The phenomenon of resonance has still greater significance in the design shown by Figure 83-c, in which the sound absorber is covered with a perforated sheet of solid material. Here the field of absorption undergoes further displacement in the low frequency range. Very great absorption in the low frequency range may be obtained with the aid of special resonator-absorbers; an outline diagram of one of these is shown in Figure 83-d. Designs in which the principle of resonant sound absorption is utilized are discussed in Paragraph 30.

The type of design of sound absorber selected depends upon the type of acoustic task to be resolved and the spectrum of the noise which is to be absorbed. Thus in the absorption of high frequency noises (which, as mentioned in the foregoing, have a very irritating

effect), an absorber of very simple type may be applied, such as a layer of absorbing material applied directly to the wall of a partition or absorbing structure (absorber, sound isolating cowling, etc.). At the same time, depressions, grooves and slits may be made in porous sound absorbing materials, such as in acoustic plasters, or fibrous absorbers, in order to improve the sound absorption.

To increase the intelligibility of speech under noisy conditions, and also for absorbing low frequency noise it is advantageous to select one of the absorber designs shown in Figure 83, b-d.

28. Reverberation in a Room. Absorption of Diffuse Sound

When the source of sound in a room suddenly ceases action, the sound does not disappear immediately, but gradually. If this after-noise is of long duration, it is said that the reverberation of the room is high. Conversely, rapid extinction of the sound corresponds to a low reverberation of the room.

The criterion of the magnitude of reverberation is the time of reverberation, i.e. the length of time during which residual sound is reduced by a certain amount. This amount is taken as 60 db, which corresponds approximately to the sound level of speech above the standard zero threshold.

Evaluation of the magnitude of the time of reverberation is of interest primarily from two points of view. Firstly, it enables precise determination of the demands relative to large rooms, from the point of view of ensuring adequate intelligibility of speech in them, or if necessary (in music halls) ensuring the quality of musical sound. Secondly, and this is more important for ship designers, the magnitude of the reverberation time enables determination of the degree of absorption of sound in rooms and evaluation of the sound absorbing materials and structures under the conditions of a realistic (diffuse) sound field.

We feel intuitively that the less the absorption of sound in a room the greater is the reverberation time, and conversely, the greater the absorption, the shorter is the time. This conclusion comes to mind involuntarily when we remember the hollowness of sound in large rooms with concrete or metallic walls, and how quickly sound dies out in rooms with an abundance of carpets and curtains. The results of theoretical and experimental investigations substantiate this original hypothesis, and indicate that the reverberation time t_{rev} is connected with a quantity A, characterizing the absorption of sound in a room, in the following manner:

$$t_{rev} = \frac{0,16V}{A}, \tag{116}$$

where V is the volume of the room in m^3; and the quantity A is equal to:

$$A = \alpha_1 s_1 + \alpha_2 s_2 + \alpha_3 s_3 + \ldots = \Sigma \alpha_i s_i. \tag{117}$$

Here $\alpha_1, \alpha_2, \ldots, \alpha_i$ are the coefficients of absorption of individual portions of partitions of the room or structures within the room;

$s_1, s_2, \ldots, s_i$ are the areas of these portions in m^2.

The unit of absorption of noise conventionally is taken as 1 m^2 of open window, which may be considered as completely absorbing the noise generated in the room. This unit of absorption is called the Sabin in the foreign literature. In countries in which the unit of measurement of area is the square foot, the Sabin is a correspondingly smaller unit. In converting absorption expressed in British or US Sabins into metric Sabins the corresponding figures must be decreased by the factor 10.75.

Still another value, the average coefficient of absorption of sound in a room, is used in acoustic computations involving rooms:

$$\alpha_{av} = \frac{A}{s} = \frac{\Sigma \alpha_i s_i}{s}, \tag{118}$$

where A is the total sound absorption of the room;
S is the total area of all internal surfaces of the room.

Formula (116) holds for values of $\alpha_{av} < 0,25$. At greater values of the average coefficient of absorption, better results are obtained with the formula:

$$t_{rev} = \frac{0,16V}{s \ln\left(\frac{1}{1-\alpha_{av}}\right)}. \tag{119}$$

A corrective member is introduced in the expression t_{rev} for rooms with very large volume, for taking into account air absorption. In view of the lack of such large rooms on shipboard this corrective member is not included in the expression t_{rev}.

Reverberation time may be determined experimentally with the aid of the apparatus utilizing a logarithmic level recorder (Figure 84).

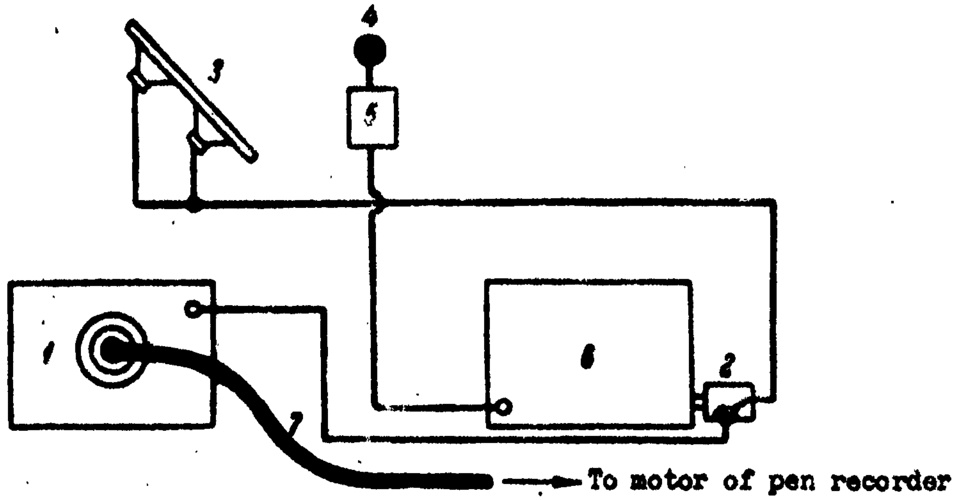


Figure 84. Very simple apparatus for automatic recording of reverberation curves.

1 - Sound generator; 2 - Interruptor on the shaft of the motor of the self-recorder; 3 - Loudspeakers; 4 - Microphone; 5 - Amplifier; 6 - Logarithmic pen recorder; 7 - Flexible shaft connecting the recorder motor with the frequency dial of the sound generator.

The motor of the recorder is joined to the frequency dial of the sound generator by a flexible shaft. The rotation of the dial is synchronized in this manner with the movement of the paper of the self-recorder. The output of the generator through the interruptor, also mounted on the cable of the recorder, is connected with the loudspeakers. Thus with the rotation of the recorder motor, the loudspeakers are periodically disengaged from the sound generator. With the aid of the corresponding microphone circuit the reverberating sound is received and, converted into an electric voltage, is fed into the input of the recorder.

The curves of reverberation at various frequencies have the form shown in Figure 85. Knowing the speed of motion of the pen recorder tape and the value of the divisions (db) on the tape, the value of t_{rev} may be computed directly. Even simpler to use is a special transparent protractor (119) on which the values of t_{rev} for various speeds of the tape and various potentiometers of the recorder are indicated. In using this protractor one of the base lines of the protractor is aligned with the reverberation curve (on a logarithmic scale it usually is a straight line) corresponding to the conditions of measurement.

The value of the reverberation time is read off a horizontal line of the scale intersecting the center of the protractor. Thus in Figure 85 the reverberation time is determined, under conditions of the use of a 50 db potentiometer in the recorder, and a tape advancement speed of 10 mm/sec (indicated on the protractor). The reverberation time in the given example is 5 seconds, which is characteristic of rooms with low sound absorption. More complex systems may be suggested, also utilizing a level recorder, with the aid of which the frequency function of the reverberation time of a room may be obtained automatically.

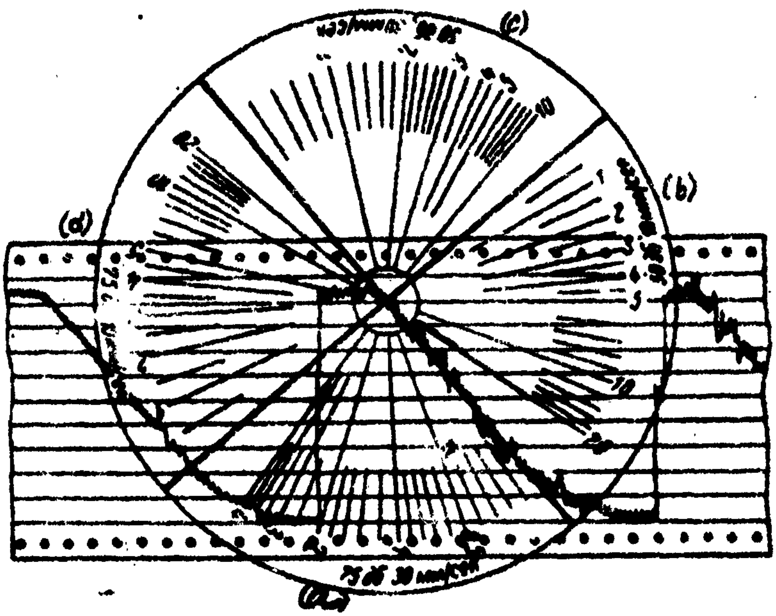


Figure 85. Examples of recordings of reverberation curves and method of measuring therefrom the reverberation time, with the aid of a protractor.
(a) 75 db 30 mm/sec; (b) 50 db 10 mm/sec; (c) 50 db 30 mm/sec; (d) 75 db 10 mm/sec.

What is the desirable level of reverberation time in rooms in which speech or music are heard? From the point of view of intelligibility of speech the optimum reverberation time may be taken as 0.5 sec for small rooms, and 0.8 to 1 sec for large rooms. High quality perception of music requires approximately a two-fold greater reverberation time. These data relate to a frequency of 500 to 1,000 cycles. At lower and higher frequencies the optimum reverberation time increases.

Let us turn now to another very important application of the reverberation method -- in determining the coefficients of sound

absorption of materials and structures in a diffuse (dispersed) sound field. The preceding paragraph introduced data on the sound-absorbing properties of materials with normal angle of incidence of sound. The coefficients of absorption of noise by the same materials in a diffuse field, i.e. under realistic conditions, most often are greater than under normal angle of incidence.

In measurement of the coefficients of noise absorption in a diffuse sound field, a reverberation chamber is used, i.e. a room with large reverberation time. When a noise absorber is introduced into a room of this type the reverberation time is reduced correspondingly. A simple mathematical derivation enables the following expression to be obtained for the value of the coefficient of absorption of the tested material:

$$\alpha = \alpha_{K} + \frac{0{,}16V}{s_{M}}\left(\frac{1}{t_{K+M}} - \frac{1}{t_{K}}\right). \tag{120}$$

Here α_K is the average coefficient of absorption of the internal surfaces of the chamber;

V and t_K are its volume in m^3, and the time of reverberation, in seconds;

S_M is the area of the sound absorbing material placed in the chamber, in m^2;

t_{K+M} is the reverberation time in the chamber after the test material is placed in it, in seconds.

For the purpose of increasing the precision of measurements in the reverberation chamber, a series of corrections are taken into account for the finiteness of the dimensions of the sample of the noise absorber, its positioning, character of the field in the chamber, etc.

Table 13 contains the coefficients of noise absorption in a diffuse field of materials and structural elements obtained by various authors (42, 56, 108). Several authors relate their data to the noise absorption of materials at frequencies of 128, 256, 512 cycles etc., and others to frequencies of 100, 250, 500 cycles, etc. The data for the corresponding frequencies have been combined in the Table, because the small difference in frequencies does not affect the magnitude of absorption.

Table 14 presents data on the noise absorption of human objects and furniture. It gives not the coefficients of absorption, but the number of units of absorption for each object. The data of Tables 13 and 14 may be utilised for computation of the sound level in rooms (cf. Chapter 8) and the reverberation time of rooms.

Table 13

Coefficient of Noise Absorption of Materials and Structural Elements at Various Frequencies

Structure or Material	Sound Frequency, cycles					
	128 (100)	256 (250)	512 (500)	1024 (1000)	2048 (2000)	4096 (4000)
Windows, Doors, Openings						
Reference structure (open window)	1	1	1	1	1	1
Closed window	0,35	0,25	0,18	0,12	0,07	0,04
Door openings	0,3	0,3	0,3	0,4	0,4	0,4
Ventilation openings	—	—	0,3—0,5	—	—	—
Walls and Overheads						
Sheet steel	—	—	0,01—0,05	—	—	—
Pine boards, 3/4"	0,1	0,1	0,1	0,06	0,08	0,1
Concrete	0,01	0,01	0,02	0,02	0,03	0,04
Plywood, 3 mm, on 5 cm beams	0,2	0,28	0,26	0,09	0,12	0,11
Plywood, 8 mm, on 5 cm beams	0,28	0,22	0,17	0,09	0,1	0,11
Plywood, 16 mm, on 4 cm beams	0,18	0,12	0,1	0,09	0,08	0,07
Wood fiber panel, 25 mm	0,18	0,11	0,19	0,39	0,95	0,56
Floors						
Floor, mastic rubbed, on wooden beams	0,15	0,11	0,1	0,07	0,06	0,07
Parquette on asphalt	0,04	0,04	0,07	0,06	0,06	0,07
Linoleum, 5 mm, on floor	0,02	—	0,03	—	0,04	—
Metlakh slabs	0,01	—	0,015	—	0,02	—
Ordinary carpet	0,09	—	0,2	—	0,27	—
Carpet, felt lined	0,11	—	0,37	—	0,27	—
Coconut mats on solid floor	0,08	—	0,17	—	0,3	—
Rubber, 5 mm, on floor	0,04	0,04	0,08	0,12	0,03	0,1
Cork, 9.5 mm, on floor	0,08	0,2	0,08	0,19	0,21	—
Curtains and Drapes						
Heavy curtain, 9 cm from wall	0,06	0,10	0,38	0,63	0,7	0,73
Cotton curtain (0.5 kg/m^2), directly against wall	0,04	0,07	0,13	0,22	0,32	0,35
Velvet cloth (0.65 kg/m^2), directly against wall	0,05	0,12	0,35	0,45	0,38	0,36

Table 13 (continued)

Structure or Material	Sound Frequency, cycles					
	128 (100)	256 (250)	512 (500)	1024 (1000)	2048 (2000)	4096 (4000)
Velvet cloth, 10 cm from wall	0,06	0,27	0,44	0,5	0,4	0,35
Same, 20 cm from wall	0,08	0,29	0,44	0,5	0,4	0,35
Canvas or linen, 15 cm from wall	0,1	0,12	0,25	0,33	0,15	0,35
Several Acoustic and Heat Insulation Materials						
Mineral wool, 25 mm	0,09	0,23	0,53	0,72	0,75	0,77
Same, 50 mm	0,2	0,53	0,74	0,78	0,75	0,77
Same, 100 mm	0,68	0,84	0,82	0,78	0,75	0,77
Construction felt, 12.5 mm ...	0,05	0,06	0,17	0,48	0,52	0,51
Same, 25 mm	0,15	0,22	0,54	0,63	0,57	0,52
Same, 50 mm	0,34	0,5	0,69	0,67	0,58	0,52
Same, 75 mm	0,5	0,66	0,77	0,68	0,58	0,52
Asbestos felt, 10 mm	0,06	0,14	0,32	0,25	0,19	—
Glass felt, 30 mm	0,06	0,12	0,36	0,81	—	—
Vermiculite	—	—	0,4	0,45	0,35	0,5
Asbestos-silicate	0,4	0,6	0,8	0,83	0,82	0,76
Aluminum wool, 40 mm	0,18	0,35	0,55	0,67	0,63	0,63
Acoustic plaster, 10 mm	—	0,03	0,07	0,11	0,2	0,34
VT-4 material (caprone fiber), 50 mm *	—	0,25	0,41	0,71	0,91	—
Same, 100 mm *	0,27	0,3	0,64	0,91	0,91	—
Asbestos floss braid, 25 mm*..	—	0,34	0,43	0,47	0,77	—
Same, 50 mm*	—	0,35	0,45	0,49	0,79	—

* According to tube measurement data.

Table 14

Sound Absorbing Properties of People and Furniture

Object	Number of units of absorption per object at frequency, cycles					
	125	250	500	1000	2000	4000
Audience on wooden chairs	0,17	0,36	0,47	0,52	0,5	0,46
Musicians (with instruments)..	0,4	0,85	1,15	1,4	1,3	1,2
Viennese chair................	0,01	0,02	0,02	0,02	0,02	0,02
Chair with plywood seat & back	—	0,01	0,01	0,02	0,04	0,05
Armchair with upholstery seat and back	0,11	0,18	0,28	0,35	0,45	0,42

Example 27. The saloon of a diesel passenger ship has the dimensions 15 x 10 x 4 m, and is used for both speech (lectures, reports) and as a concert hall. The walls and overhead have been recommended to be covered with decorated 8-mm plywood, and the floor is to be parquette. There are 50 chairs in the saloon. The total area of windows and ports is 3 m^2, and of wooden apertures is 8 m^2. The problem is to evaluate the quality of the room from the point of view of intelligibility of speech and music. What recommendations may be made for improving the reverberation properties of the room?

Solution. We compute the total sound absorption in the room and the time of reverberation at various frequencies, utilizing the data of Tables 13 and 14. The results of the computation are shown in the following table. In determining the reverberation time, at the end of the table the volume of the saloon is taken as equal to 600 m^3.

Surfaces and Objects	Area, m^2, or No.	Number of Absorption Units at Frequency, cycles					
		125	250	500	1000	2000	4000
Walls & ceiling	350	350× ×0,28=98	350× ×0,22=77	350× ×0,17=60	350× ×0,09=31	350× ×0,1=35	350× ×0,11=39
Floor..........	150	5	6	11	9	9	11
Windows.........	3	1	1	0,5	0,5	—	—
Wooden openings	8	2,5	2,5	2,5	3	3	3
Audience.......	30	5	11	14	15,5	15	14
Total sound absorption....	—	112,5	97,5	88	59	62	67
Reverberation time, sec. (formula 116)	—	0,85	1,0	1,1	1,63	1,55	1,4

Bearing in mind the foregoing relative to the value of the optimum reverberation time we find that a room having in the given case a medium magnitude of volume is much more suitable for listening to music than listening to speeches. For the purpose of increasing the intelligibility of speech the reverberation time must be reduced, i.e. absorption at medium and high sound frequencies must be increased. This may be attained, for example, by screening part of the wall with curtains or special drapes.

For the purpose of reducing the curtain area, the curtain selected must be heavy, with a large coefficient of sound absorption. Let us assume the total area of a curtain is 80 m^2. The problem is to compute the sound absorption, similarly to the above computation. The values of the coefficients of absorption of the curtain may be taken from the data of Table 13.

Surfaces and Objects	Number of Absorption Units at Frequency, cycles					
	125	250	500	1000	2000	4000
Walls and ceiling (270 m^2).....	75.5	59.5	46	24	27	30
Curtains (80 m^2)...............	5	8	30	50	56	58
Other sound-absorbing objects..	14.5	20.5	28	28	27	28
Total sound absorption.........	95	88	104	102	110	116
Reverberation time, seconds....	1.0	1.1	0.9	0.94	0.87	0.83

Upon approximate calculation the room may be considered satisfactory with respect to the established requirements. In the case of musical, and especially orchestral presentation the curtains (drapes) may be withdrawn into special openings or false columns. This increases reverberation at medium and high frequencies to a certain extent, and improves the quality of sound.

Another structural resolution of this problem may be to cover the walls and ceiling with special sound-absorbing materials instead of plywood. This reduces the reverberation time at medium sound frequencies and increases the reverberation time somewhat at 125 and 250 cycles, which is desirable in the given case.

29. Sound Field in a Room with Sound Absorption

Let us assume that a sound source, radiating sound equally in all directions, is set in a room with partially absorbing partitions. At each point of a room of this kind the intensity of sound field J may be considered as consisting of the intensity of the straight (diverging) sound J_s and the intensity J_d of sound dispersed as a result of repeated reflection from the partially absorbing borders of the room.

$$J = J_s + J_d . \tag{121}$$

The intensity of sound in watts/cm^2 at any given point of the field of a diverging spherical wave is determined by the expression (55):

$$J_s = \frac{W}{4\pi r^2} , \tag{122}$$

where W is the acoustic power of the source, also expressed in watts;

r is the distance from the source to the given point, in centimeters.

The intensity of the dispersed sound may be represented by the expression (56):

$$J_d = \frac{4W}{R} . \tag{123}$$

Here R is the so-called constant of the room, equal to:

$$R = \frac{S\alpha_{av}}{1 - \alpha_{av}} , \tag{124}$$

where s is the total area of the room partitions;
α_{av} is the above-mentioned average coefficient to absorption of sound in the room, equal to:

$$\alpha_{av} = \frac{1}{s}(s_1\alpha_1 + s_2\alpha_2 + s_3\alpha_3 + \ldots + s_n\alpha_n); \tag{125}$$

$s_1, s_2, s_3 \ldots s_n$ are the areas of individual portions of the borders of the room, having the corresponding coefficients of sound absorption $\alpha_1, \alpha_2, \alpha_3 \ldots \alpha_n$.

The total intensity of sound at any given point of the room thus will be:

$$J = W\left(\frac{1}{4\pi r^2} + \frac{4}{R}\right). \tag{126}$$

The sound level (cf. formula 65) is equal to:

$$\beta = 10 \log \left(\frac{J}{10^{-16}}\right) = \beta_W + 10 \log \left(\frac{1}{4\pi r^2} + \frac{4}{R}\right) \text{ db.} \tag{127}$$

In this expression β_W is the so-called power level of the source:

$$\beta_W = 10 \log \frac{W}{10^{-16}} \text{ db.} \tag{128}$$

Under practical conditions it is more convenient to take the distance from the sound source to the receiver in meters instead of centimeters. Formula (127) assumes the form:

$$\beta = \beta_W + 10 \log \left(\frac{1}{4\pi r_M^2} + \frac{4}{R}\right) - 40 \text{ db.} \tag{129}$$

where r_M is the distance in m.

The difference between the level of the sound intensity and the power level of the source is equal to:

$$\beta - \beta_W = 10 \log \left(\frac{1}{4\pi r_M^2} + \frac{4}{R}\right) - 40 \text{ db.} \tag{130}$$

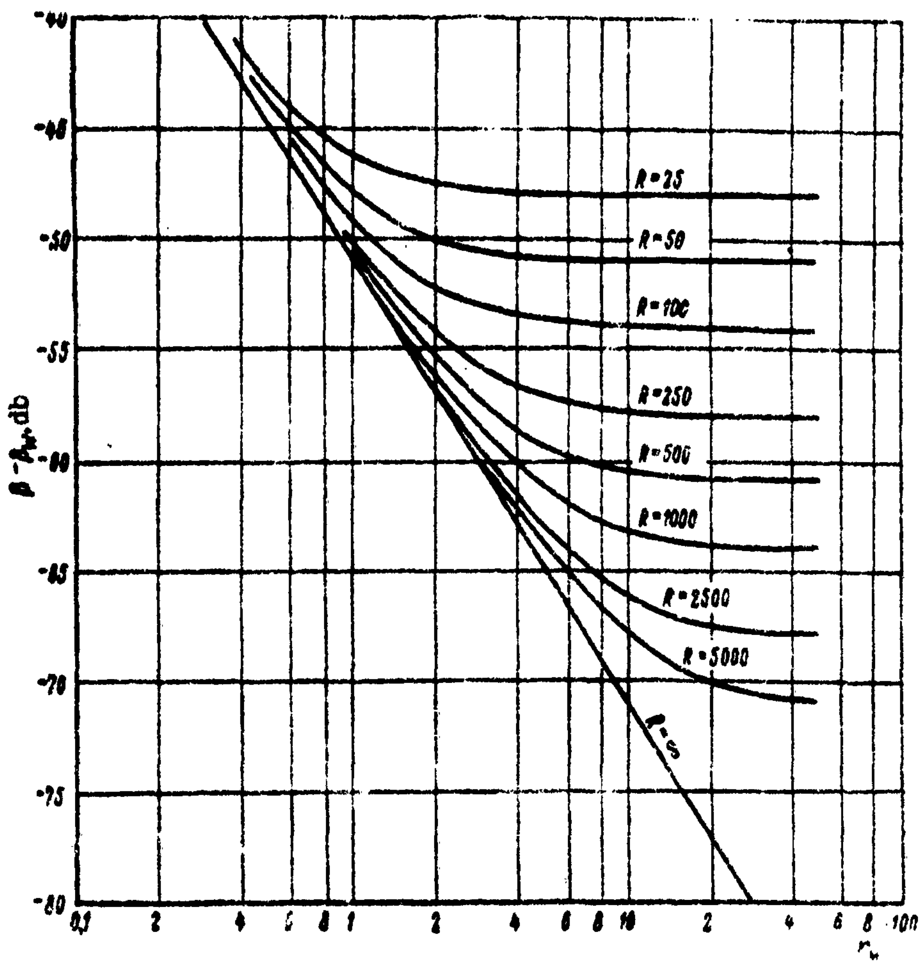


Figure 86. Difference between the level of intensity of sound in a room and power level of a non-directional source, as a function of the distance from the source (R is the constant of the room).

This value is negative because the expression under the logarithm sign is less than unity. It is plotted in Figure 86 for a series of values r_M and R. The straight line on the graph corresponds to $R = \infty$, i.e. $\alpha_{av} = 1$, which is the same as the absence of any barrier around the source. In this case, the second member in the parentheses in expressions (126) through (130) is equal to zero, i.e. there is only a diverging spherical sound wave.

The curves of Figure 86 give a visual picture of the character of the diminution of sound levels with distance in rooms of various dimensions or with various sound absorption. It is apparent that at sufficient distances from the source, the diminution in the sound level ceases. At these distances the level is determined only by sonic vibrations reflected from the borders of the room. In the construction of the curves of Figure 86 the considerable interference phenomena which occur in practice, or variations of sound levels as a result of local reflection of sound, or its partial screening, were not taken into account. The curves also do not hold in the immediate vicinity of the room boundaries. At the sites of greatest curvature of the curves, where the straight field is equal to the diffuse field, computations based on the curve are only approximate.

Example 28. A machine of small size is located on a platform in the center of a room 6 x 10 x 3.5 m. The acoustic power radiated by the machine in the course of its operation, is equal to 0.1 watt, and the average coefficient of absorption of the boundaries of the room in the given frequency range is 0.3. The problem is to determine the sound level at distances of 1 and 4 meters from the source.

Solution. The total area of the boundaries of the room is:

$$s = 2(10\cdot 6 + 10\cdot 3{,}5 + 6\cdot 3{,}5) = 232\ м^2.$$

The constant of the room is:

$$R = \frac{232\cdot 0{,}3}{1-0{,}3} = 100\ м^2,$$

The power level of the source is:

$$\beta_W = 10\log\frac{0{,}1}{10^{-16}} = 150\,\text{db}.$$

We find the difference between the levels of intensity and power at various distances is found according to the graph (Figure 86), whence we then determine the sought-for values β.

At a distance of 1 m from the source

$$\beta - \beta_W \cong -49\ \text{db};\quad \beta = 150 - 49 = 101\ \text{db}.$$

At a distance of 4 m from the source

$$\beta - \beta_W \cong -54\ \text{db};\quad \beta = 150 - 54 = 96\ \text{db}.$$

Example 29. Acoustical testing of machinery is performed in a shop the dimensions of which are 40 x 20 x 5 m. The walls of the shop are stucco (coefficient of absorption ≈ 0.1). The problem is to determine the degree of difference between the sound level measured at a distance of 1 m from the mechanism and the level when the mechanism is placed in the open air.

Solution. We determine the constant of the room:

$$s = 2200\ \text{м}^2; \quad R = \frac{2200 \cdot 0{,}1}{0{,}9} \approx 250\ \text{м}^2.$$

From Figure 86 it is apparent that at a distance of 1 m from the sound source the level at R = 250 differs from the level at $R = \infty$ is no greater than 0.5 db. Therefore the noise level of the mechanism measured in the shop is practically equal to the level when the machine is located in the open air.

The appended table refers to a small non-directional sound source located in the center of a room. When the source is located near the walls or corners the radiation is concentrated in definite directions due to reflection from the walls. The directionality of radiation may be characterized by the coefficient of concentration Q, representing the ratio of intensity of sound along the axis of a directed radiator to the intensity which would obtain in the case of the action of a non-directional source of the same power.

The values Q for various locations of a non-directional source of sound in the room are included in Table 15 (117).

Table 15

Coefficient of Directionality for Small Non-Directional Source of Sound Located at Various Places in a Rectangular Room

Location of sound source	Coefficient of directionality Q
Near the center of the room	1
At the center of a wall	2
At the intersection of two walls, at the mid-height of the room	4
In a corner of the room	8

The expression for the sound level will have the following form:

$$\beta = \beta_W + 10 \log \left(\frac{Q}{4\pi r_M^2} + \frac{4}{R} \right) - 40 \text{ db.} \tag{131}$$

At Q = 1 (non-directional radiation) the formula transforms to formula (129). The values $\beta - \beta_W$ from expression (131) are represented in Figure 87. Formula (131) and the graph may be utilized both for determination of the sound field of a non-directional source located near a wall of the room, and for computation of the field of directed loudspeakers or a system of loudspeakers in rooms with partially sound-absorbing walls.

Example 30. Under the conditions of the room described in Example 28, the problem is to determine the sound level at distances of 1 and 4 meters from a mechanism transferred to a corner of the room.

Solution. The coefficient of concentration of radiation in this case is equal to Q = 8. Because curves for R = 100 are lacking in the graph (Figure 87), we turn to formula (131).

For a distance of 1 meter we obtain:

$$\beta = 150 + 10 \log \left(\frac{8}{4\pi \cdot 1^2} + \frac{4}{100} \right) - 40 = 108 \text{ db.}$$

At a distance of 4 m the sound level is:

$$\beta = 150 + 10 \log \left(\frac{8}{4\pi \cdot 4^2} + \frac{4}{100} \right) - 40 \simeq 99 \text{ db.}$$

The levels for the same distances from the source have increased with respect to the levels of Example 28 due to the directionality of radiation.

Example 31. The noise level at a control station in an engine room is 90 db. The acoustic constant of the room is R = 50. The problem is to determine the acoustic power of a group of loudspeakers necessary to ensure 70 percent articulation of transmission of commands at a distance of 5 meters from the loudspeakers in the direction of their axes. The coefficient of concentration of radiation is 8.

Solution. From Figure 21 we find that at a noise level of 90 db the speech level necessary for ensuring 70-percent articulation must be

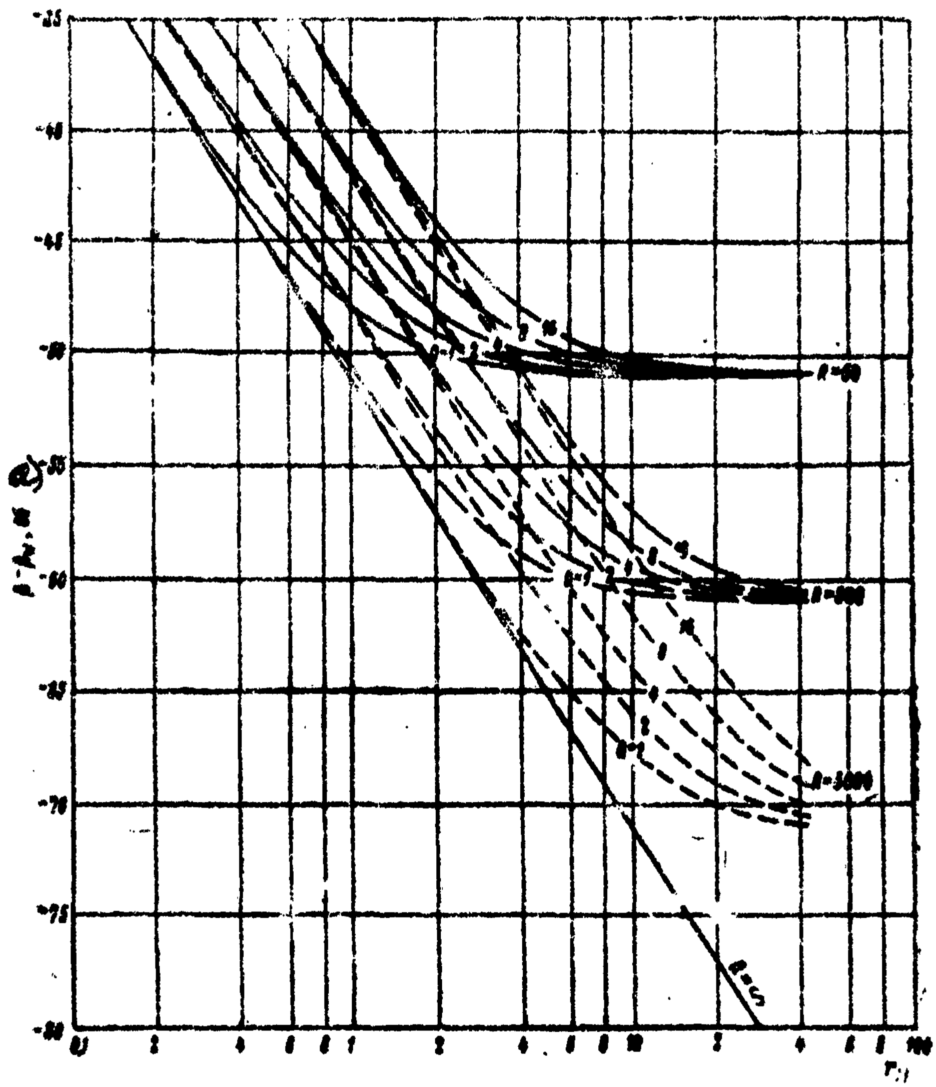


Figure 87. Same as Figure 86, but for a directed source, with coefficient of concentration of radiation Q.

a) $\beta - \beta_W$, db (ordinate); distance from source, meters (abscissa).

equal to 107 db. On the basis of formula (131) the level of acoustic power of the source (loudspeakers) is equal to:

$$\beta_W = 107 - 10 \log \left(\frac{8}{4\pi \cdot 5^2} + \frac{4}{50} \right) + 40 \cong 157 \text{ db.}$$

The acoustic power of the loudspeakers (according to formula 128) is:

$$W = 10^{-16} \cdot 10^{0,1\beta w} = 10^{-0,3} = 0,5 \text{ watt.}$$

Knowing the electroacoustic efficiency of the loudspeakers used, we may determine the required power at the terminals of the amplifier.

30. Resonance Sound Absorption

At the basis of resonance sound absorption, which has been indicated by Rayleigh (88), lies the concept of the utilization of a resonant system with high damping. Work in the fields of the theory and technique of resonance sound absorption was begun in the USSR during the decade of 1930 by S. N. Rzhevkin (91), who justly may be considered one of the pioneers of investigation in this field. Research was conducted on resonance sound absorption by V. S. Nesterov (80), K. A. Vital (29), M. S. Antsyferov (13) and others. G. D. Malyuzhinets developed the theory of layer-resonant systems with freely hung screens, and produced a method for computation of the characteristics of layered systems with the aid of impedance diagrams (73).

Let us consider the simplest air resonator, i.e. a container with rigid walls and narrow neck (Figure 88-a). When sound waves of a definite frequency fall upon this resonator, the air "stopper" in the

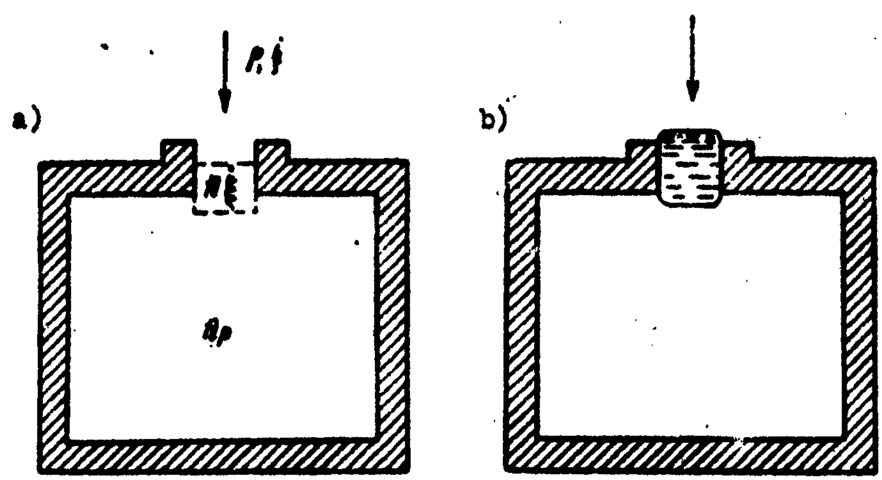


Figure 88. Resonator as a sound absorber.

neck of the container begins intense vibratory motion. The speed of vibration of the portion of air in the neck is several-fold greater

than the speed of vibration in the free sound field $\dot{\xi}$. At this time a corresponding increase in vibratory pressure p occurs in the internal volume of the resonator. If a tube is passed into the internal recess of the resonator, as done in the hearing instrument for the hard of hearing, the sound perceived aurally will appear louder.

The ordinary concept of a resonator as a sound amplifier is based on the above, although actually any resonator, including that pictured in Figure 88-a, may perform the function of a sound absorber, if the friction losses are high enough. If a layer of sound absorbing material is added in the neck of the resonator, i.e. at the site where the speed of vibration of particles is greatest (Figure 88-b), the absorbing properties of the system increase greatly at resonance. We obtain a special resonance sound absorber, utilized in construction practice. For a single resonator (Helmholtz) with resonant frequency f_0, the maximum absorption is equal to:

$$A_{max} = \frac{1}{2\pi}\left(\frac{c}{f_0}\right)^2. \tag{132}$$

where

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{S}{l_к V}} = \frac{5,4\cdot 10^3}{\sqrt{\frac{l_к V}{S}}}. \tag{132a}$$

Here c is the speed of sound in air, in cm/sec (in formula 132, m/sec);
S is the cross-sectional area of the neck of the resonator, cm^2;
V is the volume of the internal recess of the resonator, cm^3;
l_K is the equivalent value of the length (depth) of the neck, in cm; it is somewhat greater than the actual length of the neck (l). For example, for a neck with round cross-section, with radius r:

$$l_к = l + 1,57r.$$

Example 32. The problem is to determine the resonant frequency and A_{max} of a single resonator (Figure 88) having a neck radius of 0.3 cm, neck depth 0.8 cm, and dimensions of elastic volume 10 x 10 x 5 cm.

Solution. The equivalent length of the neck is:

$$l_к = 0{,}8 + 1{,}57 \cdot 0{,}3 = 1{,}27\ см;$$

$$f_0 = \frac{5{,}4 \cdot 10^3}{\sqrt{\frac{1{,}27 \cdot 10 \cdot 10 \cdot 5}{\pi (0{,}3)^2}}} = 115 \text{ cycles;}$$

$$A_{max} = \frac{1}{2\pi}\left(\frac{340}{115}\right)^2 = 1{,}4\ м^2.$$

Single resonators are able to produce considerable sound absorption only within a narrow range of frequencies in the region of resonance. A much wider range of frequency of absorption may be obtained through application of resonance absorbers with perforated panels. S. N. Rzhevkin, V. S. Nesterov and other authors developed a series of practical designs of absorbers of this type and methods for computation of their characteristics (92). Of these designs the small-size resonance-panel ZP-4 absorber may be considered for application in ships. It consists of a thin, perforated metallic or plastic composition sheet 0.5 mm thick (perforation pitch 17 mm, diameter of perforations 5 mm), fastened 80 mm from the wall. A cloth with fairly large coefficient of friction, approximately 12 to 15 dynes sec/cm, is glued to the underside of the sheet. The values of the coefficients of friction of several materials are shown in Table 16, taken from the aforementioned work of S. N. Rzhevkin and V. S. Nesterov. This absorber provides fairly good absorption $(\alpha > 0{,}5)$ within a frequency range of 400 to 4,000 cycles.

Table 16

Coefficient of Friction of Fibrous Materials

Material	Coefficient of friction $\frac{\text{dynes} \cdot \text{second}}{\text{centimeter}}$
Gauze	0.4 - 0.5
Thin calico, technical	1
Thin cheap cotton cloth	3
Dense metallic screen (grid 0.12 mm, wire thickness 0.05 mm)	3
Loose calico	5 - 6
Baize	40 - 80
Various glass fiber cloths	3 - 100

Table 17 presents the values of coefficients of absorption of structures with perforated screens, based on the data of the Moscow Radio Transmission Directorate (42). The absorption of these designs depends to a considerable extent upon the resonance effect.

Table 17

Name of Structure or Material	Sound frequency, cycles						
	125	250	500	1000	2000	4000	6000
Perforated panels: (Filler - asbestos wadding)							
δ=3 мм; d=4 мм; D=40 мм; l=50 мм.	0,27	0,43	0,36	0,25	0,15	0,13	0,11
δ=3 мм; d=4 мм; D=40 мм; l=100 мм	0,47	0,47	0,36	0,28	0,25	0,27	0,28
δ=3 мм; d=6 мм; D=25 мм; l=50 мм	0,20	0,46	0,58	0,52	0,42	0,31	0,31
δ=3 мм; d=6 мм; D=25 мм; l=100 мм	0,52	0,54	0,54	0,50	0,41	0,33	0,33
δ=3 мм; d=7 мм; D=30 мм; l=50 мм	0,19	0,36	0,45	0,43	0,30	0,24	0,22
δ=3 мм; d=7 мм; D=30 мм; l=100 мм	0,45	0,51	0,55	0,48	0,34	0,21	0,17
δ=6 мм; d=4 мм; D=40 мм; l=50 мм	0,32	0,42	0,31	0,18	0,13	0,10	—
Wood fiber slabs of the Orgalit type, 11 mm thick, vol. wt. 200-250 kg/m^2:							
mounted directly on wall...	0,06	0,15	0,28	0,30	0,33	0,31	0,35
5 cm from wall.............	0,22	0,30	0,34	0,32	0,41	0,42	0,47
10-mm plywood panels, 10 cm from wall.........	0,34	0,19	0,10	0,09	0,12	0,11	0,06
Same, 3-mm plywood...........	0,32	0,35	0,19	0,13	0,11	0,10	0,08

Notes: δ - thickness of plywood; d - diameter of perforations; D - distance between perforation centers; l - thickness of filler layer.

During recent years a resonance absorber of the slit type has acquired extensive application. It is distinguished from the Helmholz resonator by the presence of a long slit at the opening, the length of which corresponds to the length of the internal air cavity. The expression for the frequency of maximum absorption of the slit resonator has the form (42):

$$f_{os} = \frac{c}{2\pi}\sqrt{\frac{b}{S_p}} \cdot \frac{1}{\sqrt{l + \frac{2b}{\pi}\left(1{,}12 + \ln\frac{c}{\pi b f_{os}}\right)}} =$$

$$= \frac{5{,}4 \cdot 10^3}{\sqrt{\frac{S_p l}{b}}} \cdot \frac{1}{\sqrt{1 + \frac{2b}{\pi l}\left(1{,}12 + \ln\frac{c}{\pi b f_{os}}\right)}},$$

where b is the width of the slit, cm;
l is its depth, cm;
S_p is the cross-sectional area of the air cavity perpendicular to the length of the slit, cm^2.

The first factor is the resonance frequency, computed as though the slit resonator were a Helmholz resonator. This may be readily seen by multiplying the numerator and denominator of the radicand by the length of the slit (which is the length of the plates) and comparing the expression thus obtained with formula (132a). The second (a correction) factor includes the sought-for quantity f_{os}, and because of this, computation of the actual value of f_{os} must be performed by the method of successive approximations.

Example 33. The problem is to determine the average frequency of the range of effective absorption of a slit resonator having the following dimensions: slit width b = 0.3 cm, slit depth l = 2 cm; dimensions of the cross-section of the air cavity corresponding to each slit, 3 x 4 cm.

Solution. Determining the value of the first factor in the frequency expression, we have:

$$\frac{5{,}4 \cdot 10^3}{\sqrt{\frac{3 \cdot 4 \cdot 2}{0{,}3}}} = 600 \text{ cycles.}$$

Computing the value of the frequency of a slit resonator, taking the second (correction) factor into account, we have, for the first approximation:

$$f_{os}^1 = 600\,\frac{1}{\sqrt{1 + \frac{2}{\pi}\cdot\frac{0{,}3}{2}\left(1{,}12 + 2{,}3\lg\frac{34000}{\pi \cdot 0{,}3 \cdot 600}\right)}} = 490 \text{ cycles.}$$

The value of the frequency in the second approximation is computed in similar manner:

$$f_{o2}^{2} = 600 \frac{1}{\sqrt{1+\frac{2}{\pi}\cdot\frac{0,3}{2}\left(1,12+2,3\lg\frac{34000}{\pi\cdot 0,3\cdot 490}\right)}} \approx 485 \text{ cycles.}$$

If necessary further approximations may be computed, although comparison of the second approximation with the first shows that no marked change in frequency may be anticipated.

Therefore the average frequency of the working zone of the given slit absorber is approximately 500 cycles.

Figure 89 through 92 show several sound absorbing designs of the resonance type. Figure 89 shows slit resonators. A slit resonator (Figure 89-a) for concert halls consists of a series of profiled lathes, behind which is a sound absorber P (42). The form of the canals between the lathes is such that the absorber cannot be seen and a decorative effect is produced.

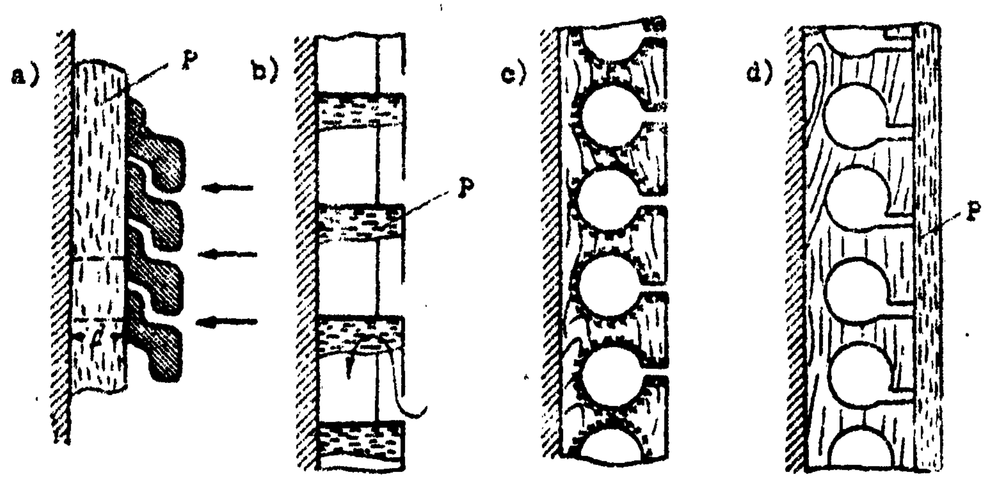


Figure 89. Slit resonators (P - absorber).

In the design shown in Figure 89-b the absorber is located only in the necks of the resonators, i.e. in places where the speed of vibration of air particles is greatest.

The "Deveton" absorber produced by German firms (Figure 89-c) is made of wood waste and has internal cylindrical cavities which are

connected with the external surface by slits. The thickness of the absorber plate is 25 to 35 mm, and its volume weight is approximately 400 kg/m^3. The coefficient of sound absorption in a diffuse field ranges from 0.2 at a frequency of 100 cycles to 0.5 at 800 cycles, and up to 1 at frequencies above 3,000 cycles. Apparently the acoustic property of "Deveton" may be improved somewhat by gluing a porous absorber to its surface. This prevents reflection from the solid external surfaces of the plates and increases the coefficient of loss of the slit resonators (Figure 89-d), although the external appearance of the absorber is worsened. Increasing the depth of the slits of the absorber lowers its frequency and improves sound absorption somewhat at low frequencies.

The double resonance sound absorber developed by C. Gilford and N. Druce (134) is of great interest. The absorber is intended for radio studios, although it may be applied in other rooms requiring high absorption. The external surface of the absorber is impermeable to sound and consists of a thin sheet of polyethylene or bitumenized felt, which in combination with a layer of mineral wool behind it forms a high frequency resonance system (Figure 90). Behind these is another resonance system with perforated screen, having a much lower natural frequency. A double resonance absorber of this type ensures a coefficient of sound absorption greater than 0.5 in the frequency range 500 to 5,000 cycles. The absence of slits or openings in the external layer of the absorber and the presence of an elastic film give the entire structure a pleasing external appearance, protects the absorber from pollution and parasite colonies, and enables the surface to be washed periodically.

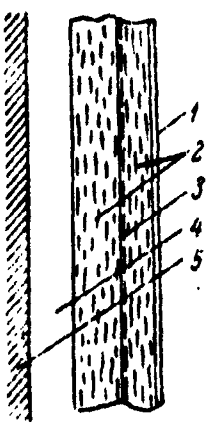


Fig. 90. Double resonance sound absorber. 1 - Polyethylene film; 2 - Sound absorbing material; 3 - Perforated metal screen; 4 - Air gap; 5 - Partition.

The aforementioned authors also developed a highly effective triple resonance sound absorbing system, although in view of its relatively large thickness (20 to 30 cm) it may hardly be utilized on ships.

Very light, cheap and rapidly installed membrane resonance sound absorbers of polyethylene (Figure 91-a) are suitable for shipbuilding practice. The range of frequencies of absorption is determined by the dimensions of the rectangular cavities of the absorber. The absorbers are fastened to the partition in a manner such that the groups of cavities are distributed in

in checkerboard fashion (Figure 91-b). The high coefficient of internal losses in the material of the absorber excludes the possibility of random amplifications of sound, which sometimes take place with membrane absorbers made of plywood.

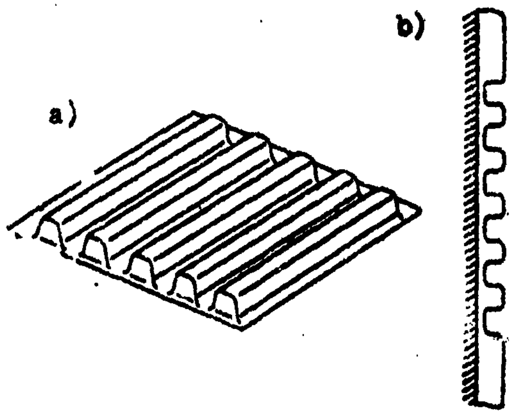


Fig. 91. Sections of membrane absorber made of polyethylene (a), and method of placing the section on the wall (b).

Of great technical advantage, especially for shipboard use, are composite absorbers, consisting of thin perforated cassettes of plastic composition or metal, filled with fibrous material. The units are fastened to special strips or directly to the wall (Figure 92-a) with the aid of simple stamped parts. The length of this type of cassette unit, produced by a Danish firm, attains 7 meters. It appears that a supplemental effect of slit resonance absorption may be acquired by attaching the cassettes away from the wall and separated slightly from each other (Figure 92-b).

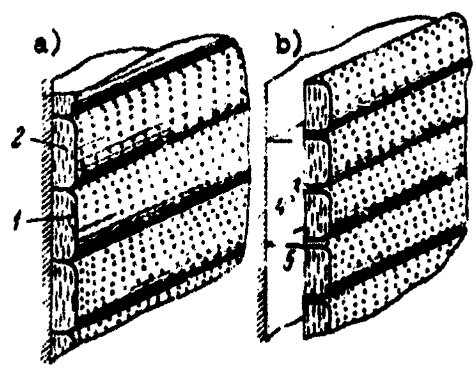


Fig. 92. Sound absorber of the cassette-panel type.
1 - Perforated metal cassettes;
2 - Sound absorbing material;
3 - Slits between cassettes; 4 - Air cavities; 5 - Connecting plates.

Resonance absorption (1) also may be obtained from sound absorbers suspended in space (cf. Figure 116). A volume resonance absorber of this type consists of a box with perforated walls and several internal partitions (Figure 93-a). This forms a series of different volumes, and the field of frequencies of absorption is increased. For the purpose of increasing the degree of absorption, the perforated walls of the box are covered with cloth. From Figure 93-b it is apparent how the introduction of the cloth, having a resistance of 21.5 dyne sec/cm, influences the coefficient of absorption of the resonator. If the distance between the individual absorbers

is less than three-fold the dimension of the absorber, their absorption is reduced as a result of mutual interaction.

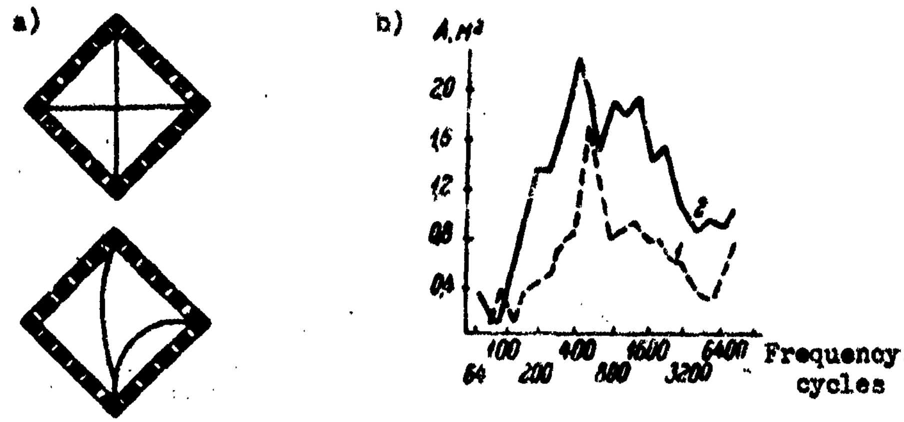


Figure 93. Volumetric multiple resonance sound absorber (a) and frequency characteristics of its absorption (b). 1 - Without cloth; 2 - With cloth.

Although resonance sound absorption is one of the relatively little developed fields of structural acoustics, and the application of resonance absorbers in specific cases involves more or less complex research work, there are increasingly definite possibilities for utilization of absorbers of this type on ships, in compartments such as saloons, radio rooms and engine rooms.

CHAPTER 8

STRUCTURAL ARRANGEMENTS FOR SOUND ISOLATION AND

SOUND ABSORPTION

31. Penetration of Sound Into Compartments Having Different Boundary Structures

Let us determine the noise level in a compartment bounded by a series of boundary structures with various sound isolation values ZI_1, ZI_2 ... ZI_n. In general, the noise levels $\beta_1, \beta_2 \ldots \beta_n$, acting on each boundary also are different (Figure 94). Within the room which is to be isolated are a series of sound absorbers with coefficients of absorption $\alpha_I, \alpha_{II} \ldots \alpha_i$, distributed over the areas $S_I, S_{II} \ldots S_i$. Let us assume that the noise level in the compartment being isolated is determined solely by airborne noise in the adjacent compartments, i.e. the level of sonic vibration transmitted to the walls of the compartment from machinery is relatively low. Let us assume also, that the sound field at the boundaries of the noisy room is a diffuse field, which is true at a fairly large distance from the sound source.

If a plane wave (i.e. a direct wave from the source) were to act on the boundary of the isolated room from the side of one of the noisy rooms, the sound isolation would be equal to:

$$ZI_1' = 10 \log \frac{J_1}{J_{tr_1}},$$

where J_1 and J_{tr_1} are the intensity of sound impinging on, and transmitted through the partition, respectively.

As is known from the foregoing (cf. example 20), the sound pressure is doubled when a plane wave impinges on a partition at an angle normal to its surface (the acoustic resistance of the medium is considerably less than the resistance of the partition). If the sound field acting on the partition is a diffuse field, the pressure is not doubled because all directions of the vector of Umov, i.e. energy flow, are of equal probability. Therefore, taking the intensity of the

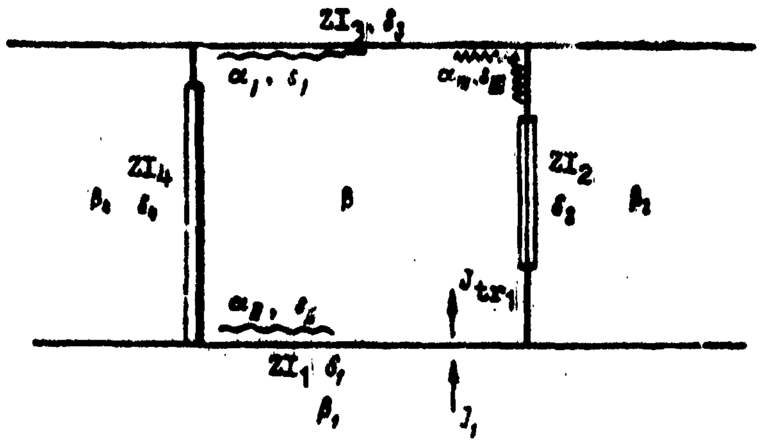


Figure 94. Determining the actual sound isolation of a compartment.

diffuse field J_{dif_1} in the noisy room into account, the sound isolation is equal to:

$$ZI_1 = 10 \log \frac{J_{dif_1}}{J_{tr_1}} = 10 \log \frac{J_1}{4J_{tr_1}} ,$$

whence

$$J_{tr_1} = \frac{J_1 \cdot 10^{-0.1ZI_1}}{4} .$$

The sound power transmitted into the room through all boundaries is:

$$W_{tr} = S_1J_1 \cdot 10^{-0.1ZI_1} + S_2J_2 \cdot 10^{-0.1ZI_2} + S_3J_3 \cdot 10^{-0.1ZI_3} + \ldots + S_nJ_n \cdot 10^{-0.1ZI_n} ,$$

where S_1, S_2, S_3 ... S_n are the areas of the boundaries.

From formula (126) and the curves of Figure 86, it is apparent that at a sufficient distance from the source, when the intensity of sound is determined solely by the dispersion of sound, $J = \frac{4W}{R}$.

In the center of the room which is being isolated the field also is diffuse, and thus its power is equal to $W = \frac{JR}{4}$.

Under the usual conditions the coefficient of absorption of sound rarely exceeds 0.3 or 0.4 in the room. In these cases the constant R of the room (formula 124) may, without great error, be taken as equal to:

$$R = \alpha_{av} S,$$

or, if the room contains many sound absorbers with coefficients of absorption α_i and areas s_i,

$$R = \sum \alpha_i s_i.$$

At large values of α_{av}, precise computation of R according to formula (124) is necessary.

Equating the values of sound energy W_{tr} penetrating into the room and the energy W disseminated in it, and dividing both parts of the equation by the threshold strength of sound J_o, we obtain:

$$S_1 \frac{J_1}{J_0} 10^{-0.1ZI_1} + S_2 \frac{J_2}{J_0} 10^{-0.1ZI_2} + S_3 \frac{J_3}{J_0} 10^{-0.1ZI_3} + \ldots + S_n \frac{J_n}{J_0} 10^{-0.1ZI_n} = \frac{J}{J_0} \sum \alpha_i s_i.$$

It may be noted that:

$$\frac{J_1}{J_0} = 10^{0.1\beta_1}$$

$$\cdots\cdots$$

$$\frac{J_n}{J_0} = 10^{0.1\beta_n}.$$

Taking the logarithm of both parts of the equation:

$$10 \log \left[S_1 10^{0.1(\beta_1 - ZI_1)} + S_2 10^{0.1(\beta_2 - ZI_2)} + S_3 10^{0.1(\beta_3 - ZI_3)} + \ldots + S_n 10^{0.1(\beta_n - ZI_n)}\right] = 10 \log \frac{J}{J_0} + 10 \log \left(\sum \alpha_i s_i\right).$$

The first member of the right-hand side of the equation is no other than the noise level inside the room, β. Finally, we obtain:

$$\beta = 10 \log \left[S_1 10^{0.1(\beta_1 - ZI_1)} + S_2 10^{0.1(\beta_2 - ZI_2)} + S_3 10^{0.1(\beta_3 - ZI_3)} + \ldots + \right.$$
$$\left. + S_n 10^{0.1(\beta_n - ZI_n)}\right] - 10 \log \left(\sum a_i s_i\right) \text{ db.} \qquad (133)$$

From this general formula we may obtain formulas corresponding to particular cases. If the noise level acting on all the various walls is the same:

$$\beta_1 = \beta_2 = \beta_3 = \ldots = \beta_0 ,$$

then the noise level in the isolated room is:

$$\beta = \beta_n + 10 \log \left[S_1 10^{-0.1 ZI_1} + S_2 10^{-0.1 ZI_2} + S_3 10^{-0.1 ZI_3} + \right.$$
$$\left. + \ldots + S_n 10^{-0.1 ZI_n}\right] - 10 \log \left(\sum a_i s_i\right) \text{ db.} \qquad (134)$$

With identical levels of external noise it is advantageous to utilise the concept of average actual sound insulation of a room $ZI_{F.av}$, which is equal to the difference in noise levels outside and inside the room:

$$ZI_{F.av} = \beta_0 - \beta = -10 \log \left[S_1 \cdot 10^{-0.1 ZI_1} + S_2 \cdot 10^{-0.1 ZI_2} + \right.$$
$$\left. + S_3 \cdot 10^{-0.1 ZI_3} + \ldots + S_n \cdot 10^{-0.1 ZI_n}\right] + 10 \log \left(\sum a_i s_i\right) \text{ db.} \quad (135)$$

The following particular case is one in which the sound isolation of all boundaries is identical: $ZI_1 = ZI_2 = \ldots = ZI_n = ZI$.

From formula (134) we obtain:

$$\beta = \beta_0 - ZI + 10 \log \left(\frac{\Sigma S_i}{\Sigma a_i s_i}\right) \text{ db.} \qquad (136)$$

where $\sum S_i$ is the total area of the boundaries upon which sound acts from the outside.

The actual sound isolation of the room is:

$$ZI_F = ZI - 10 \log \left(\frac{\Sigma S_i}{\Sigma \alpha_i S_i} \right) \text{ db.} \qquad (137)$$

Here, as in the formulas presented in the foregoing, the value of the sound isolation, ZI, for different types of partitions is taken from Tables 8, 9 and 10.

Let us assume that an absorber of a single type, characterized by coefficient of absorption α, is located in the isolated room. If the area related to this absorber is equal to the area of the partition through which the sound passes, the sound level in the isolated room will be:

$$\beta = \beta_0 - ZI - 10 \log \alpha \text{ db,} \qquad (138)$$

and the actual sound isolation is:

$$ZI_F = ZI + 10 \log \alpha \text{ db.} \qquad (139)$$

At $\alpha = 1$, $ZI_F = ZI$, i.e. the sound isolation effect of the barrier is fully realized. As $\alpha \to 0$ formula (138) gives the value $\beta = \infty$ for the noise level in the isolated room (because $\log 0 = -\infty$). However, even in this idealized case the level will not increase without limit because backwards radiation of noise occurs through the barrier. The reverse flow of sound is equivalent to the presence in the isolated room of a sound absorber with coefficient of absorption α', equal to the ratio of the intensity of sound J_{tr} penetrating through the barrier, to the intensity of the incident sound J_1, i.e. in the given case:

$$10 \log \alpha' = 10 \log \frac{J_{tr}}{J_1} = -10 \log \frac{J_1}{J_{tr}} \quad .$$

The above expression was presented at the beginning of the present paragraph, and is none other than the sound isolation of the boundary. Thus we obtain:

$$\beta = \beta_0; \quad ZI_F = 0, \qquad (140)$$

i.e. without the sound absorber the sound isolating boundary does not fulfill its role.

Let us determine the value of a change in the noise level in a sound-isolated compartment when the magnitude of sound absorption is changed in it. With an average coefficient of absorption α_1, the level is equal to (formula 138):

$$\beta_1 = \beta_0 - ZI - 10 \log \alpha_1 \text{ db};$$

the same holds true for a coefficient of absorption α_2:

$$\beta_2 = \beta_0 - ZI - 10 \log \alpha_2 \text{ db}.$$

The change in level in the room is:

$$\Delta\beta = \beta_1 - \beta_2 = 10 \log \frac{\alpha_2}{\alpha_1} \text{ db}. \tag{141}$$

As may be seen from formula (139), the change in value of the average actual sound isolation of the compartment can be expressed in an analogous way.

Example 34. The coefficient of absorption of steel plates is 0.01 to 0.05 (Table 13). However, under actual shipboard conditions, as a result of painting, the absorption of sound by instruments and equipment, and other factors, the average coefficient of absorption in a compartment with steel bulkheads is somewhat greater, and may be taken as 0.1. The problem is to determine the change in sound level in a compartment when sound absorbers with average coefficient of absorption of 0.4 are placed on the bulkheads.

Solution. Let the average coefficient of absorption of the bulkheads equal 0.1. The change in sound level in the room as a result of the acoustic treatment of the walls will be:

$$\Delta\beta = 10 \log \frac{0.4}{0.1} = 6 \text{ db}.$$

It may be noted that a value of 6 - 7 db is close to the maximum value obtained under practical conditions as a result of the installation of sound absorbers.

In computing the sound levels and sound isolation values according to formulas (133) - (135) we encounter computation of terms of the form:

$$C_i = S_i \cdot 10^{0.1(\beta_i - ZI_i)}; \quad D_i = S_i \cdot 10^{-0.1 ZI_i}.$$

Computation of these terms for a series of frequencies consumes a considerable amount of time. It may be suggested that a nomographic chart be used, which considerably shortens the mathematical process (Figure 95). The values $\beta_i - ZI_i$ and ZI_i are plotted on the right

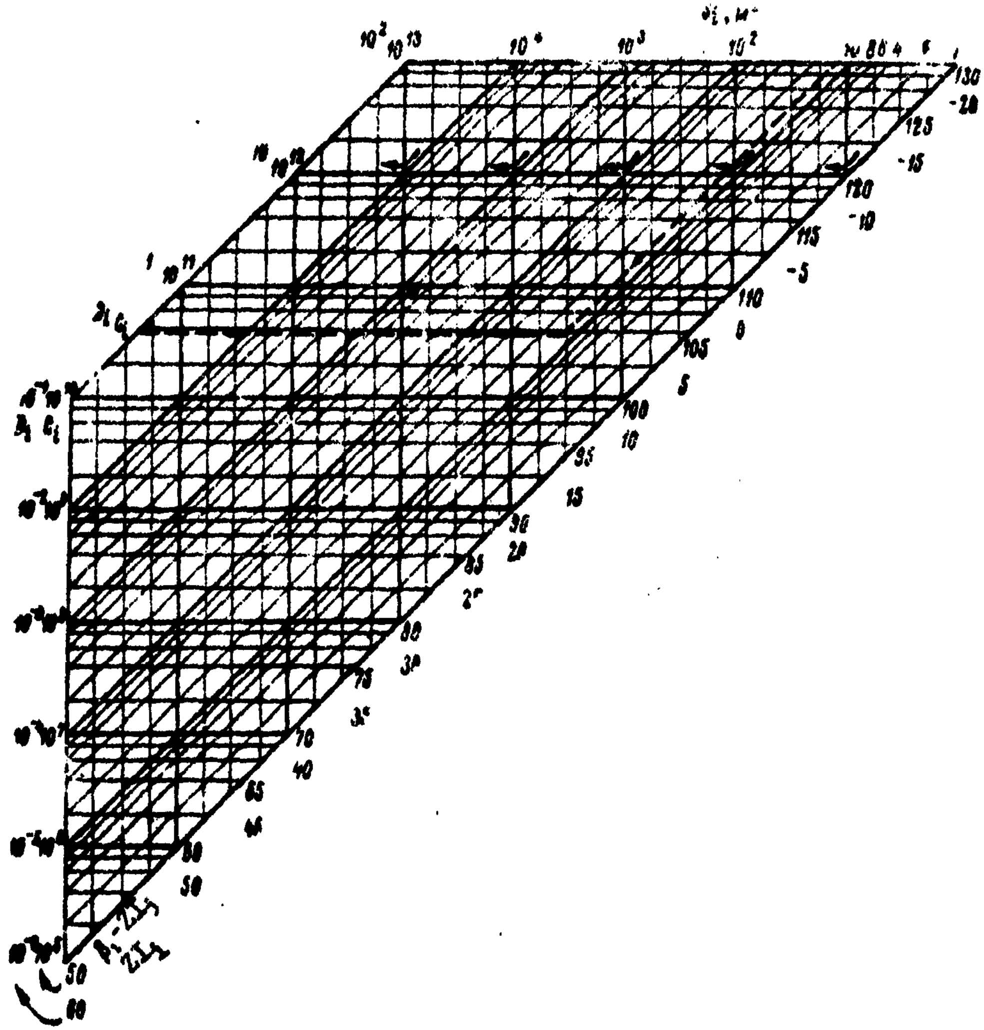


Figure 95. Nomograph for determining the values $C_i = S_i \cdot 10^{0.1(\beta_i - ZI_i)}$ and $D_i = S_i \cdot 10^{-0.1ZI_i}$, included in the expression for actual sound isolation.

hand inclined scale of the nomograph chart. The values S_i in square meters are shown along the top of the chart, and the values C_i and D_i which are sought may be read off the left side of the nomograph, on the horizontal line lying at the intersection of the lines joining the values of S_i and β_i - ZI, or ZI_i. The nomograph also enables the reverse computation to be performed, i.e. given the values of the summations $\sum C_i$ and $\sum D_i$, entering at the left of the chart, the values of

$10 \log \sum C_i$ and $- 10 \log \sum D_i$ included in the expressions (133) - (135) may be found. For this purpose, the right-hand portion of the lower scale, with large values of $\beta_i - ZI_1$ and negative values of ZI_1, is used.

Example 35. The areas of the walls of a room are 12, 40 and 20 m^2, and the sound isolation values of these walls are 15, 30 and 20 db, respectively. The noise level outside the first two walls is 110 db, and the level of noise acting on the third wall is 95 db. The problem is to determine the level of noise penetrating into the room if the total sound absorption inside the room is 100 units.

Solution. According to the computed differences $\beta_i - ZI_1$ we find the values of C_1 by means of the nomograph (Figure 95). Following computation of $\sum C_i$, by means of the same nomograph we find $10 \log \sum C_i$, after which, taking into account corrections for sound absorption, we determine the sought-for noise level. The computation is presented in the following table. An example of the determination of C_1 for $S_1 = 12\ m^2$ is indicated by dotted lines in Figure 95.

S_i	β_i	ZI_1	$\beta_i - ZI_1$	C_i
12 40 20	110 110 95	15 30 20	95 80 75	$4 \cdot 10^{10}$ $4 \cdot 10^{9}$ $6 \cdot 10^{8}$
			ΣC_i	$\sim 4{,}5 \cdot 10^{10}$

Entering with this value of $\sum C_1$ at the left scale, we plot a horizontal line to the right-hand scale ($S = 1$). This gives the value of $10 \log \sum C_i$, in the given case ~ 107 db. The noise level inside the room (formula 133) then is:

$$\beta = 107 - 10 \log \cdot 100 = 87 \text{ db.}$$

Example 36. Under the conditions of the preceding example, the problem is to determine the average sound isolation and noise level of a room if the noise level acting on all three walls of the room are identical, and equal to 110 db.

Solution. Similarly to the above, we determine the value D_i by means of the nomograph.

S_i	ZI_i	D_i
12 40 20	15 30 20	$4\cdot10^{-1}$ $4\cdot10^{-2}$ $2\cdot10^{-1}$
	ΣD_i	$6{,}4\cdot10^{-1}$

By means of the nomograph we find that $-10 \log \sum D_i \cong 2$ db. The average actual sound isolation of the room (formula (135) then is:

$$ZI_{F.av} = 2 + 10 \log \cdot 100 = 22 \text{ db}.$$

The noise level within the room is:

$$\beta = 110 - 22 = 88 \text{ db}.$$

In cases in which the number of walls with different sound isolation does not exceed two, the computation of the sound isolation of the room may be simplified. Figure 9. resents a graph for determining the average values of sound isolation of two walls, characterized by sound isolation values ZI_1 and ZI_2 ($ZI_1 > ZI_2$). The difference between the values ZI_1 and ZI_2 are shown on the vertical scale. The parameter for the curved lines is the percentage of area occupied by the wall having the lesser sound isolation. The average sound isolation of the two walls is found by adding to ZI_1 a correction, determined on the upper horizontal scale. (The correction has negative values).

The average sound isolation value obtained characterizes the maximum value of sound isolation (without taking the actual absorption in the room into account). To obtain the true or actual sound isolation, as before a correction is made for sound absorption. With the use of the graph loss of sound isolation due to doors, windows and openings with reduced sound isolation also may be determined.

Example 37. The sound isolation of the walls, floor and ceiling of a room is $ZI_1 = 40$ db, the sound isolation of the doors and windows is $ZI_2 = 25$ db, and the doors and windows occupy $\Delta S = 20$ percent of

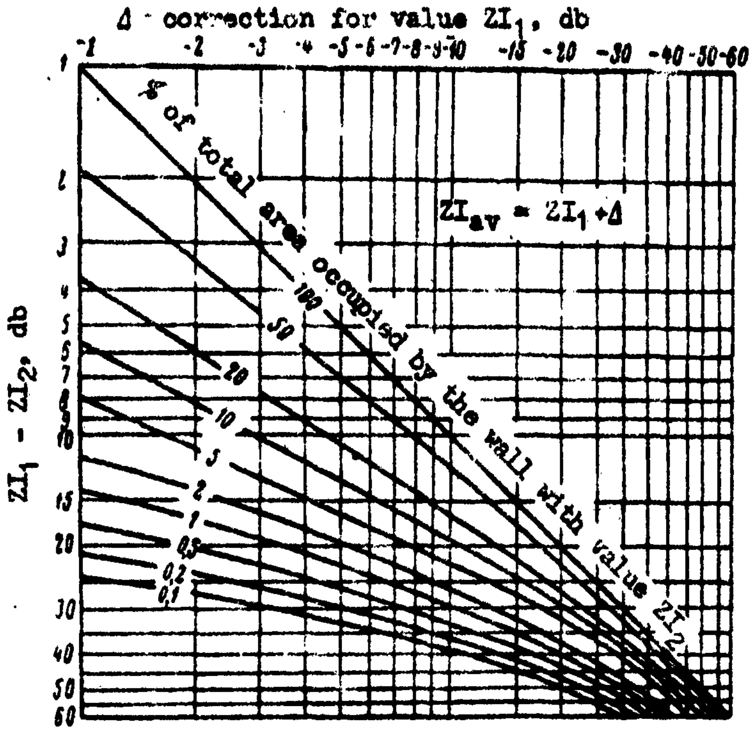


Figure 96. Average sound isolation of two walls with different sound insulation values.

the total area of the walls, floor and ceiling. The problem is to determine the average sound isolation of the walls of the room. By what amount is the average sound isolation reduced if ZI_2 is decreased to 15 db?

Solution. The difference in sound isolation values ZI_1 and ZI_2 is:

$$ZI_1 - ZI_2 = 40 - 25 = 15 \text{ db.}$$

We find according to Figure 96 that at ΔS = 20% and at the given difference in sound isolation values, the correction to the value ZI_1 is equal to - 9 db. The average sound isolation is:

$$ZI_{av} = 40 - 9 = 31 \text{ db.}$$

If the sound isolation of the doors and windows were not to exceed 15 db, then:

$$ZI_1 - ZI_2 = 40 - 15 = 25 \text{ db},$$

and the correction, still at $\Delta S = 20\%$, would attain -18 db, i.e. the isolation of the room would not exceed:

$$ZI_{av} = 40 - 18 = 22 \text{ db}.$$

Taking into account incomplete absorption of sound in the room, ZI_{av} will be still lower, i.e. it will approximate the sound isolation of the doors and windows.

32. Sound Isolating and Sound Absorbing Structures in Ships

Figure 97 shows various designs of sound isolating walls and bulkheads, categorized into six groups. The simplest designs, 1a and 1b are ordinary single steel bulkheads, decks, overheads and partitions of ships. Their sound isolation is determined by the weight per unit of surface area. Bulkhead stiffeners (design 1 b), introduced for strength, do not improve, but reduce the sound isolation of such bulkheads (Figure 70). The application of heat insulation material by gluing to the bulkhead increases the sound isolation of the latter to a very limited degree. This increase attains values of 2 or 3 db at medium sound frequencies, and 4 or 5 db at high frequencies; this results from the damping action of the heat-insulating material, and from a slight increase in mass of the bulkhead due to the addition of this material.

In the case of the designs of the first group, the deteriorating sound isolation effect due to resonance of the bulkhead can appear, consisting of resonance of coincidence and resonance of membrane-vibration of the bulkhead as a whole and of its individual parts.

The sound isolation of the second group is greater than that of the first group, due to the additional mass of the second partition, and the sound-isolating effect of the air space. The second partition (zashivka [jacket]) is made of a compact, nonporous material, such as wood, plastic composition or duralumin. The jacket is attached to the main bulkhead by means of wooden beams or studs, or which is much better, by means of elastic vibration-isolating suspension brackets made of sheet steel (design 2 c). The introduction of sound-absorbing matting between the partitions (design 2 b) reduces the resonance of the air space, and to a lesser degree reduces low-frequency resonance of the bulkhead as a whole, as a system composed of two masses connected by the elasticity of the air space.

Structures of the third group also have great sound isolation properties. In these structures the jacket consists of a layered

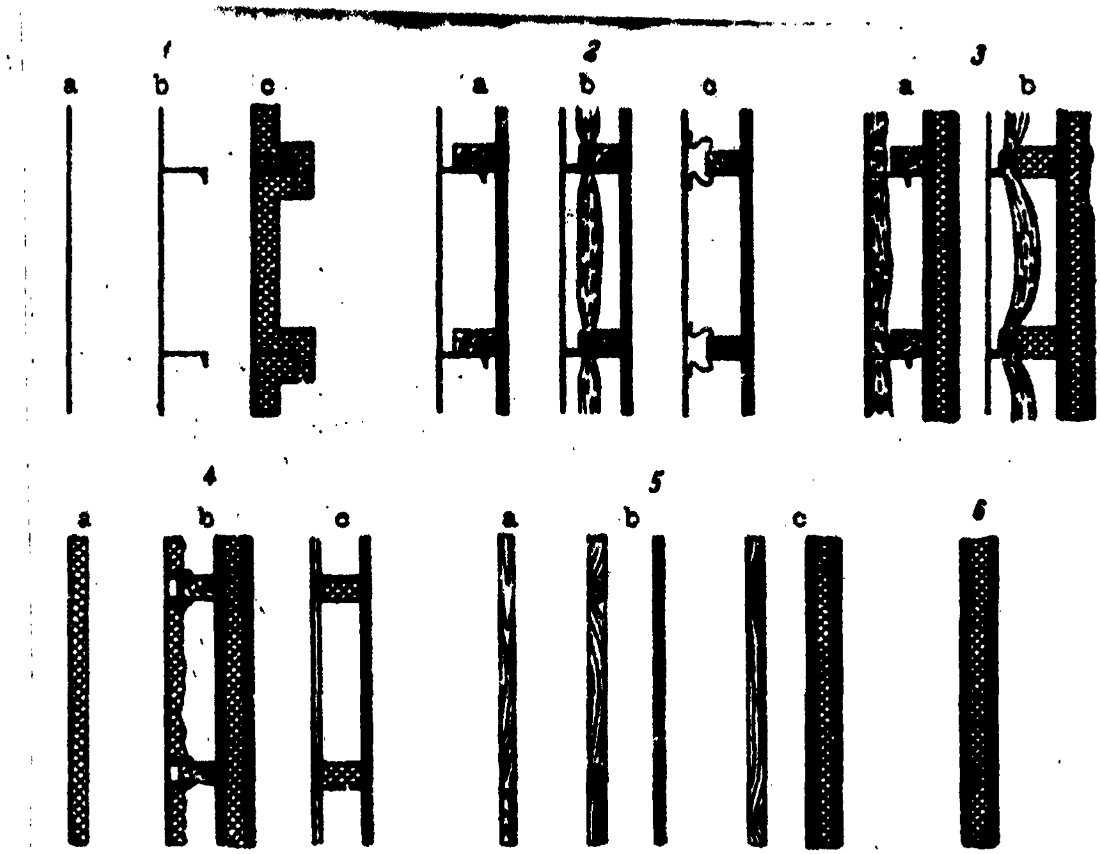


Figure 97. Sound isolating bulkheads showing structural details.

package, composed of two sheets of compact material (such as plywood) enclosing a heat insulating material between them, such as ekspanzit, asbestos-vermiculite, etc. Table 18, developed from the experimental data of E. I. Avferonk and N. R. Chetyrkin (2), indicates the degree to which very simple structures of the second and third group increase the sound isolation of single metal bulkheads. At frequencies above 1,000 cycles this increase is 9 or 10 db or more, i.e. the loudness of sound in an isolated room is reduced by more than two-fold in comparison to the effect produced by a single steel bulkhead.

This, however, does not exhaust the advantages of structures with air gaps. Only such bulkheads can isolate the vibrations caused by the sonic vibrations of metal bulkheads. Because of this, double sound isolating structures must be applied in cases in which the structure-borne noise is high, such as in small ships or in areas adjacent to the engine room, bearing in mind the fact that the

acoustic effect of single partitions is insignificant in these cases. As mentioned earlier (Figure 75-b), in the latter cases a second partition (jacket) with especially high sound isolation is desirable.

Table 18

The Increase of Sound Isolation of Bulkheads with Jackets Over the Sound Isolation of Single Bulkheads

Design group (Fig. 97)	Air gap, mm	Material and Design of jacket	Sound isolation increment, db, at frequency, cycles					
			70	140	280	560	1120	2440
2	30	Plywood, 4 mm	0	0	0	0	1	3
	60	" 4 mm	0	0	0	3	4	6
	90	" 4 mm	0	1	4	7	9	11
	90	" 15 mm	0	3	5	8	10	12
3	90	Plywood, 4 mm, heat insulating material, 20-30 mm, plywood, 4 mm	2	4	6	8	10	12

The application of helical springs for joining the elements of a double partition is ideal in principle, and amongst those available are the single-turn vibration isolation mounts type AKPO, which are used in shipbuilding. When a large air gap exists between the partitions it is very advantageous to install a layer of sound absorber between them, because a wide air gap causes the appearance of wave resonances at very low frequencies, i.e. covering a fairly wide range of frequencies.

The fourth group of structures shown in Figure 97 is distinguished from the foregoing by the introduction of substances which damp the vibration of the main metal bulkhead. This is either a vibration-damping coating (cf. Para. 54) or a second metal sheet fastened to the main bulkhead and creating a high surface friction (Fig. 71). The damping covering, if it is not very thick, may be placed on either side of the metal wall (design 4 b). If the covering is thick, it is advantageous to place it on the outside of the metal wall in double-wall structures, in order not to reduce the effective width of the air space. In as far as possible, this also should be taken into consideration in applying heat insulating material to metallic bulkheads. Use of the fourth group of designs is recommended where intense sonic vibration is encountered.

In the fifth group of designs we have included single-wall and double-wall wooden bulkheads, which also are used in ships, and the sixth group includes damping layered bulkheads of the "sandwich" type.

Figure 98 shows the design of a shipboard sound-isolating "floating" floor. The floor consists of layers of felt and asphalt, separated by oiled paper. Above these layers, on a thin layer of <u>ekspanzit</u> or cork, tongue-and-groove pine boards are laid on floor studs. According to German shipbuilding experience this type of floor completely isolates footsteps and noise arising as a result of deck vibrations.

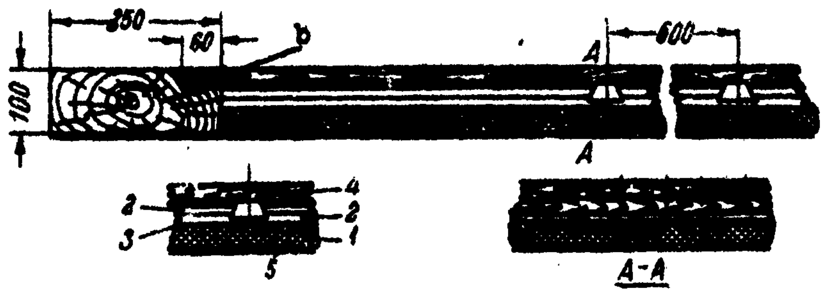


Figure 98. Design of a sound-absorbing "floating" floor.
1 - Coconut fiber felt (40 mm); 2 - Asphalt (15 mm); 3 - Oil paper; 4 - Pine boards (25 mm); 5 - Metal deck; 6 - Cork (3 mm).

Layered floors with asbestos cement slabs, and floors with springy supports also have satisfactory characteristics (cf. Table 11). In all cases, covering the floors of habited rooms of ships with carpets yields good results.

The structure of sound-isolated shipboard doors, windows, ports and hatches have been discussed in Paragraph 24. A favorable condition from the point of view of acoustics is the necessity of providing these fittings with watertightness due to shipboard conditions. This also provides the necessary sound-isolating properties (Figure 99-a). Elastic seals at the points at which bulkheads are secured (Figure 99, b-d) simultaneously provide a tight seal and prevent the transmission of sound and vibration.

Figures 100 and 101 present examples of errors frequently made when attempting noise reduction treatments on ships. In Figure 100-a a sound absorbing layer is covered over on the side of sound incidence by a solid decorative sheet, with the result that the effect of the

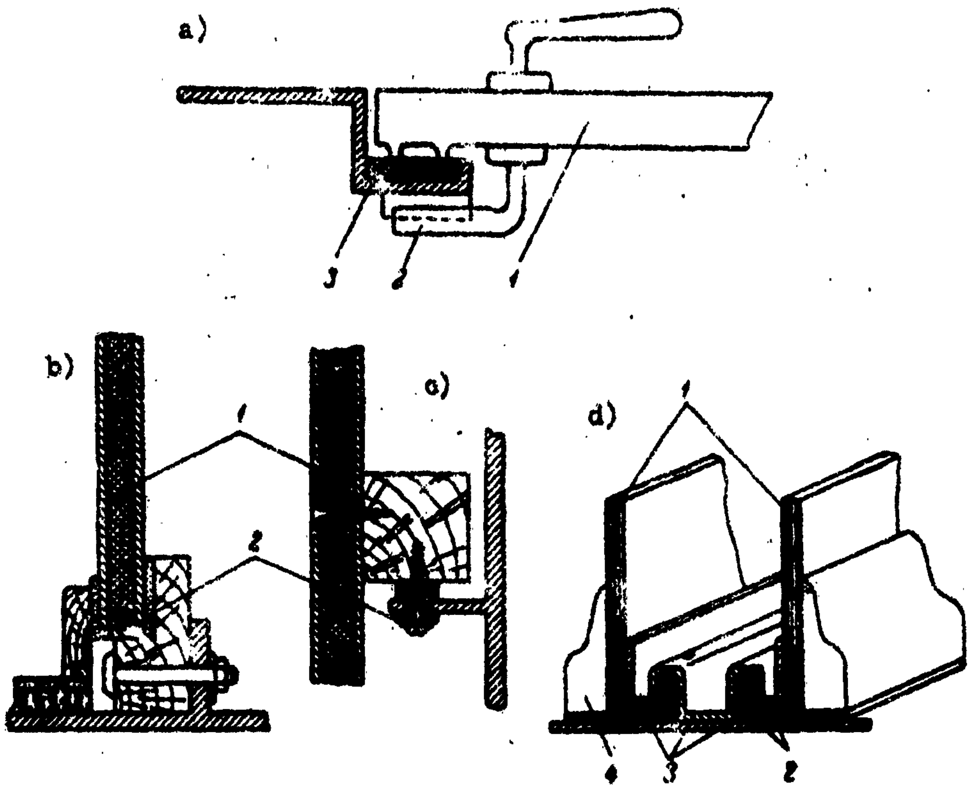


Figure 99. Details of sound-isolating watertight seal of doors and hatches (a), and method of sound-isolation securing of single-, and double-bulkheads (b, c, d).

Joint a: 1 - Stop of door or hatch cover; 2 - Wedge stopper; 3 - Rubber seal. Joints b, c, and d: 1 - Bulkheads; 2 - Cork or rubber sheet seal; 3 - Securing clamps; 4 - Skirting.

sound absorbing structure is reduced to zero. As mentioned repeatedly in the foregoing, a sound absorber may be covered only with perforated sheeting of solid metal or a metal screen (Figure 100-a, at right). It is possible, as is sometimes done, in fact, to displace the perforated sheet from the surface of the absorber by a small amount, so that the absorber will not be clogged with paint when the sheeting is painted.

In the design shown in Figure 100-b, the sound absorber is displaced from the wall. In this case a solid coating need not be applied under the sound-absorber layer because this would eliminate the supplementary effect of the air gap on the absorption of sound, and would increase the frequency at which its effective absorption begins.

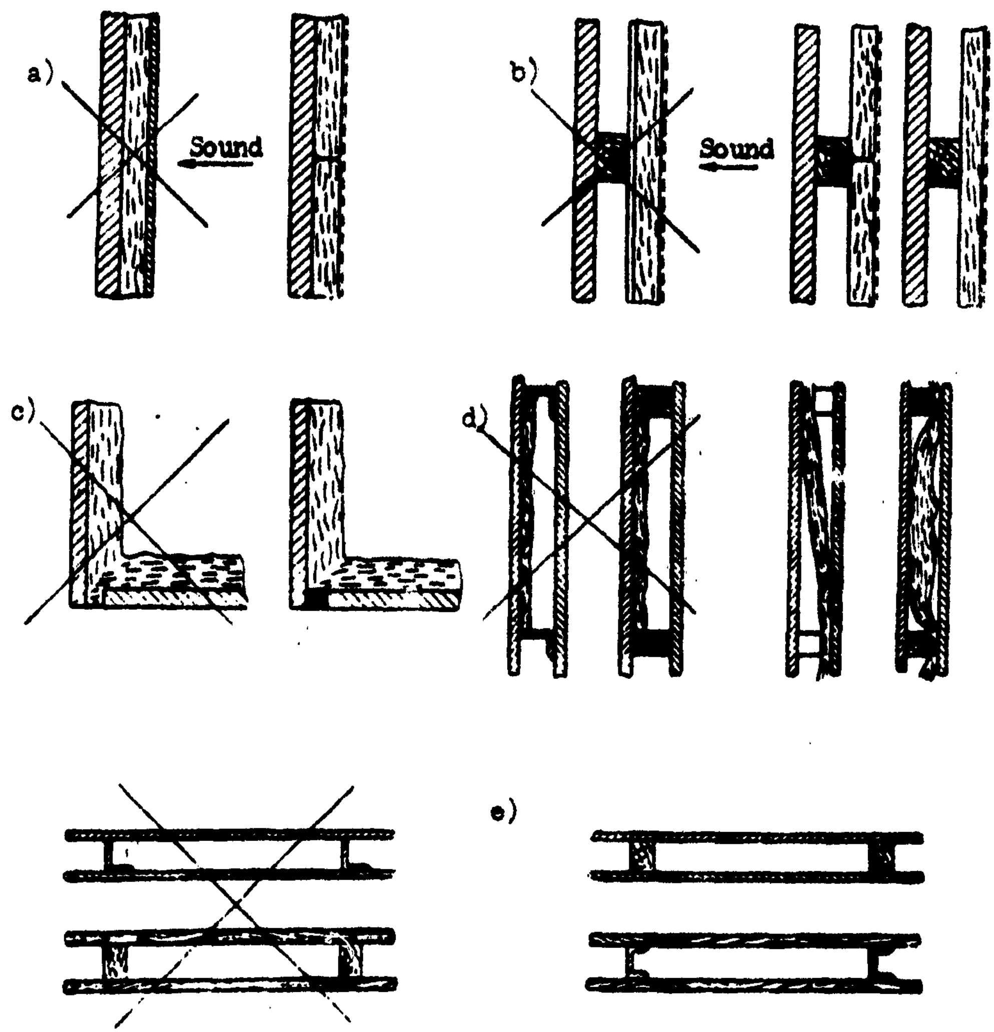


Figure 100. Examples of incorrect (crossed out) and correct design of sound absorbing and sound isolating structures.

Figure 100-c (left) shows a gap along the edge of the sound isolating structure, left during erection, which has been filled with sound absorbing material, on the assumption that this will be sufficient to prevent loss of sound isolation. Irrespective of the presence of a sound absorbing layer, the gap should be eliminated by welding up the gap, hammering material into it, or at the very least, by putting it up with a dense mastic.

Figure 100-d shows errors in the execution of a double-wall sound isolating bulkhead. In the designs shown at the left in the

figure, a dense sound absorber is located on one of the walls, but the walls are rigidly secured to each other. The sound absorber should be loose or porous, and in so far as possible should be located in the center of the air gap, where the speed of vibration of air particles, and thus sound absorption, are the greatest. Studs of rubber and soft plastic, or elastic supports are very good as connective elements. If wooden studs are used, a layer of absorber should be placed under them.

The problem of the connection of individual plates also arises in the design of decks and flooring. If the floor is not provided with special sound-absorbing characteristics (Figure 98) and is a simple double floor (Figure 100-e), the connective elements should not be made of the same material as the main flats of the floor, in order to ensure disparity of impedance of the materials, and thus reduction of sound transmission. Although at first glance it may appear somewhat strange, but in a double wooden floor the transmission of sound through the steel channel bar is less than through wooden studs (Figure 100-e, right).

Figure 101 shows several structural errors in structural arrangements which are due to a failure to take into account the transmission of sonic vibrations. The source of sound in a compartment adjacent to a machine may be not only the vibrations of the foundation of the machine, but also vibration transmitted through pipes and propeller shafting (Figure 101-a). Flexible sound isolating joints (sleeves, clutches) must be installed in the latter, and elastic supports, or as mentioned earlier, double bulkheads should be installed for isolation of the neighboring compartments (Figure 101-a, right).

The high degree of sound isolation obtained with the aid of a double-walled structure may be considerably lessened by metal plates and platforms passing through the wall (Figure 101-b). In this case the sound is transmitted to the neighboring compartment by flexural vibrations of the plates (fourth type of sound transmission, Figure 94). The plates must terminate at the boundary of one wall and, when necessary, must be "unconnected" from the boundary structures through the use of sound isolating washers and vibration absorbers.

The metal bench shown at the left in Figure 101-c, reduces the acoustic effect of a double, absorbing floor because it is rigidly secured to the vibrating metal deck, and radiates sound into the room. It is advantageous to fasten the bench to the upper, isolated portion of the floor.

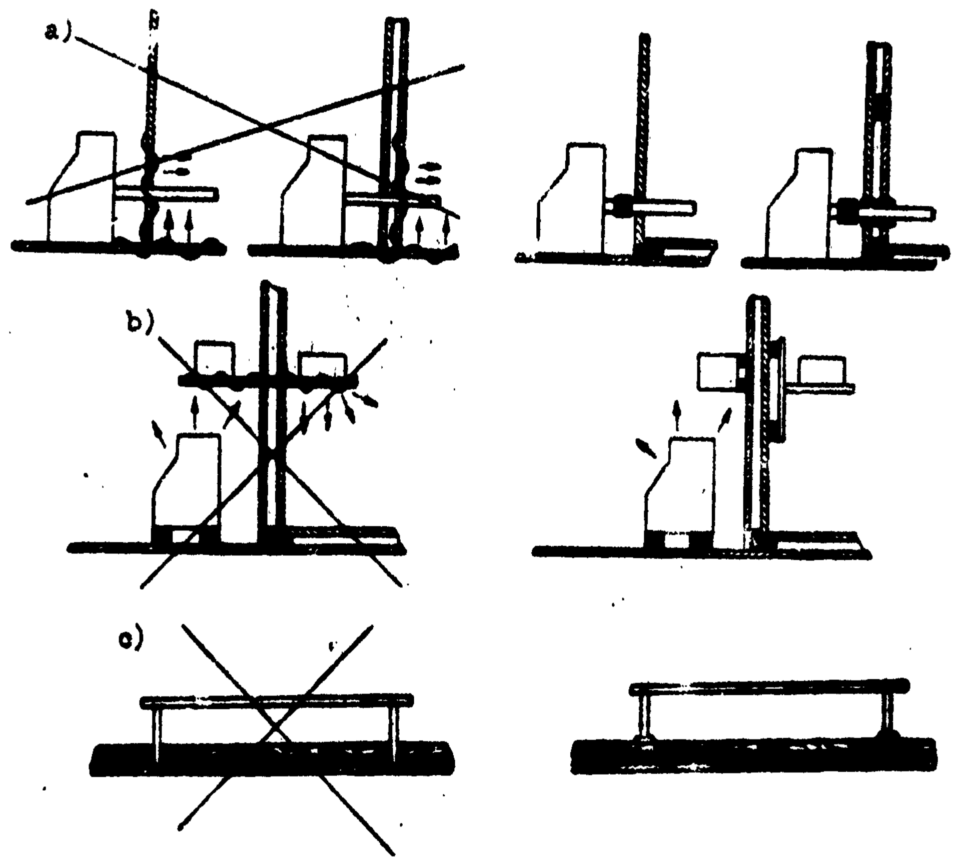


Figure 101. Incorrect and correct design of noise reducing structures subjected to intense sonic vibration.

33. Measurement of the Sound Isolation of Structures in Special Chambers and on Ships

For measuring the sound isolation of specimens of walls, bulkheads, coverings and ceilings, echo chambers, or reverberation chambers are used, i.e. rooms with low sound absorption. The arrangement of chambers of this type is shown in Figure 102. The walls of the chambers usually are smooth, made of concrete, and have low coefficients of absorption. Due to this fact the room constant R (formula 124) is low, and the field in the chamber is determined by dispersed or scattered sound (formula 126). For the purpose of further increasing the uniformity of the field, the sound sources are placed at angles to the reflecting surfaces of the room, and scatterers are employed. The walls of the chamber in which the sound sources are placed (high level chamber) are executed in a double design to avoid transmission of sonic

vibration. The sound isolating structure under test should have fairly large dimensions (2 x 2 m, and larger) for the purpose of simulating the actual structure.

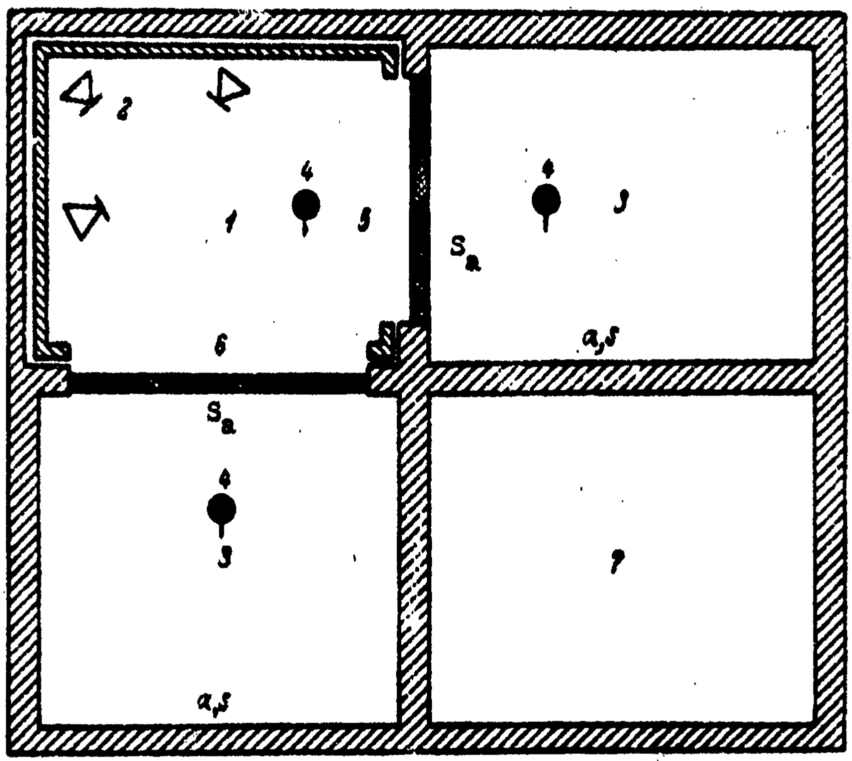


Figure 102. Arrangement of reverberation chambers for the measurement of the sound isolation of walls and similar structures.
1 - High-level chamber with loudspeakers 2; 3 - Low-level chambers; 4 - Microphones; 5 - Bulkhead under test; 6 - Decking under test; 7 - Instrument and apparatus room.

The value of the sound isolation of the structure at various frequencies is determined by the difference in sound levels in the high, and the low level chambers. From expression (137) it may be found that the actual sound isolation of the structure is:

$$ZI = ZI_F + 10 \log \frac{S_a}{a_s} \text{ db}, \qquad (142)$$

where ZI_F is the actual, or measured sound isolation, equal to the difference in sound levels in rooms 1 and 3;
S_a is the area of the sound isolating structure;

αS is the total sound absorption in the low-level chamber.

If the condition

$$S_a \approx \alpha s,$$

were to be realized, the actual sound isolation would be equal to the measured value. However, the acoustic properties of the structure being tested also affect the average coefficient of sound absorption of the chamber, and because of this and other reasons the above condition cannot be realized for different tested structures, and all frequencies of measurement. Usually, the condition

$$\alpha s < S_a, \tag{143}$$

actually exists, and the second term of formula (142) is a relatively small reverberation correction, which must be added to the measured value of sound isolation to obtain its actual value.

At large values of sound absorption the second term is negative, which contradicts the physical nature of the process and is explainable by the simplifying propositions used in the development of formula (142). In this case, according to London (21), measurements may be made in the low-level chamber, placing the microphone in close proximity to the surface of the sound isolating structure. The value of sound isolation is equal to:

$$ZI = \beta_1 - \beta_2 + \Delta p \text{ db}; \tag{144}$$

β_1 and β_2 are the sound levels of the high-, and low-level chambers respectively;

Δp is a correction, which is a function of the magnitude of $\frac{\alpha s}{S_a}$, and is determined for various frequencies from Fig. 103.

This method may be applied particularly in the approximate determination of the sound isolation of structures directly in buildings and in ships.

It may be noted that in the last-named measurements there are at least two circumstances which may distort the result to greater or lesser degree. The first is the presence of by-pass routes for the sound, such as decks, sides of the hull, and overheads, and the second is the relatively small area of the measured structure (such as a door), in comparison to the remainder of area with known sound conduction (bulkheads subdividing the room in which the source and receiver of

sound are placed). In view of these circumstances measurement of sound isolation by London's method, i.e. placing the microphone near the tested structure, must be considered as especially suitable for ships (with suitable averaging of a series of points for the elimination of any possible interference phenomena).

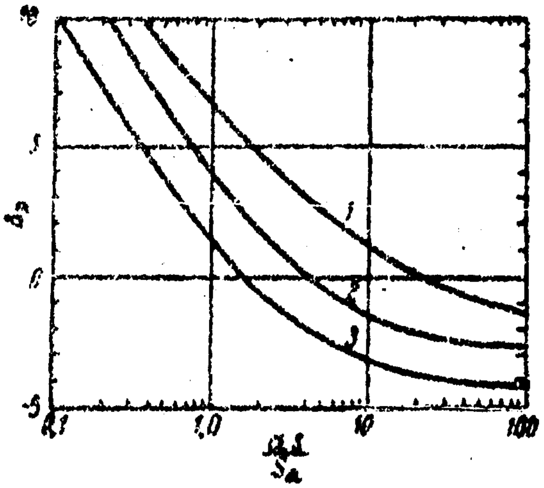


Figure 103. Reverberation correction for various frequencies when measuring sound isolation in compartments with relatively large values of sound absorption (receiving microphone placed near the sound-isolating structure).
1 - Frequency 100-200 cycles; 2 - Frequency 200-2,000 cycles; 3 - Frequency 4,000 cycles.

However, since by-pass routes influence the practical sound isolating quality of structures to a certain extent, measurement of the sound isolation of these structures "in pure form" in echo chambers does not determine completely the effect of the structure in a ship. Because of this, in addition to measuring the sound insulation of structures in a reverberation chamber, it is advantageous to conduct an evaluation of the effect of the same structure under conditions approximating reality, i.e. in chambers simulating ships' compartments.

Measurement of the sound isolation of full-size structures in chambers requires considerable expenditures of time and money. In recent years I. G. Leyzer (67) in the Soviet Union developed the theory and methodology of the model testing of sound isolation in small structural models. Application of model testing to investigation of shipboard bulkheads encounters a certain amount of difficulty because the model bulkheads are too thin. Additional research is necessary along this line.

Diagrams of the process of measurement of the sound isolation of structures, utilizing portable apparatus, are shown in Figures 104-106. This apparatus is very suitable for making measurements on shipboard, and also may be used in stationary situations. The system described includes a logarithmic level recorder, which greatly accelerates the process of measurement and provides a satisfactory and precise documentation of the readings.

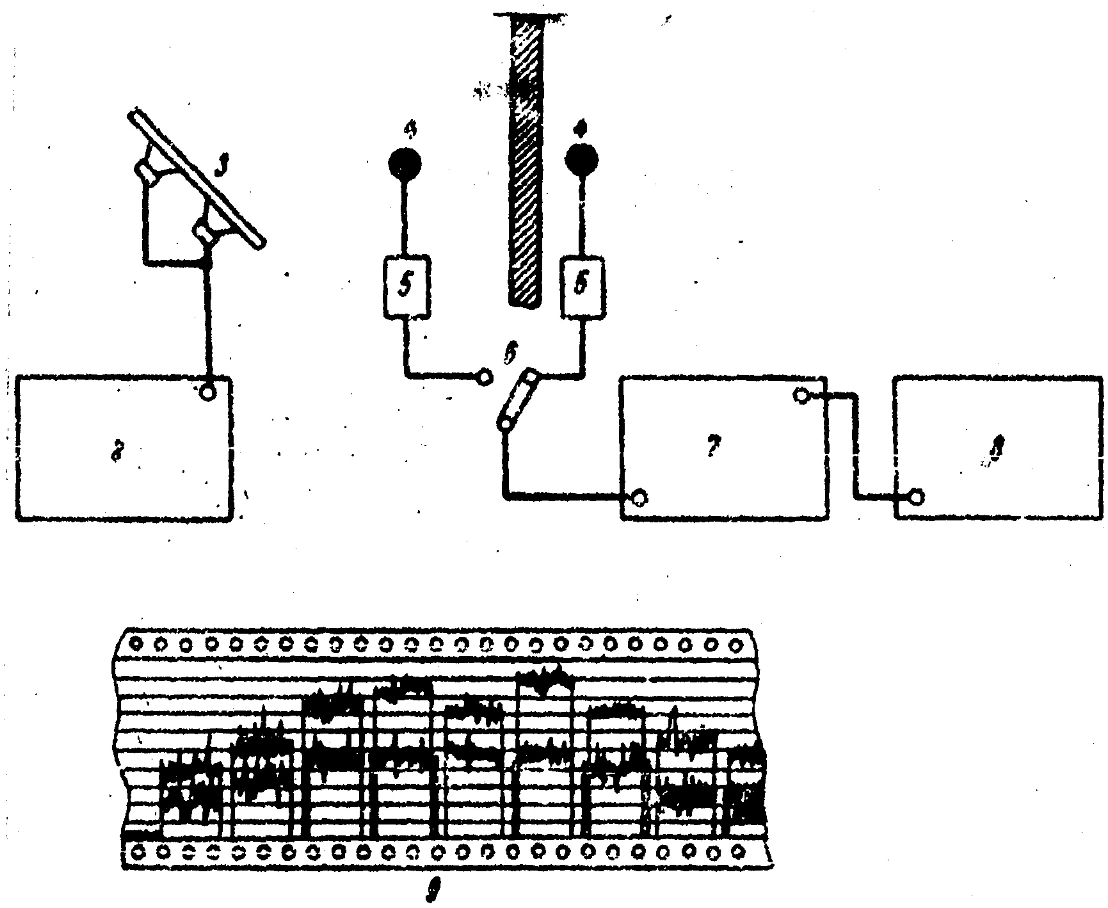


Figure 104. Arrangement for the measurement of sound isolation by frequency bands with the aid of a logarithmic intensity level recorder.
1 - The structure under test; 2 - White sound generator; 3 - Loudspeakers; 4 - Microphones; 5 - Preamplifier; 6 - Switch; 7 - Filters; 8 - Level recorder; 9 - Sample of record on the tape of the pen recorder.

In the system depicted in Figure 104 measurement is performed in the presence of white noise. The recorder records the levels in individual frequency bands (semi-octaves, third-octaves) beginning at one side of the sound isolating structure, in the "high-level" room, and then along the other side. The filters used are special sets of filters, or the filter circuit of a spectrometer. The magnitude of sound isolation in the various frequency bands is found by taking the difference in levels in these bands (taking into account the reverberation correction).

The system diagrammed in Figure 105 differs from the above by the presence of a tonal generator and a contact device on the shaft of

the recorder. With the aid of this device the outputs of microphones along both sides of the sound isolating structure are fed alternately to the recorder. The shaft of the self-recorder also is connected to the frequency dial of the sound generator, with the result that the change in frequency of sound is synchronized with the motion of the tape.

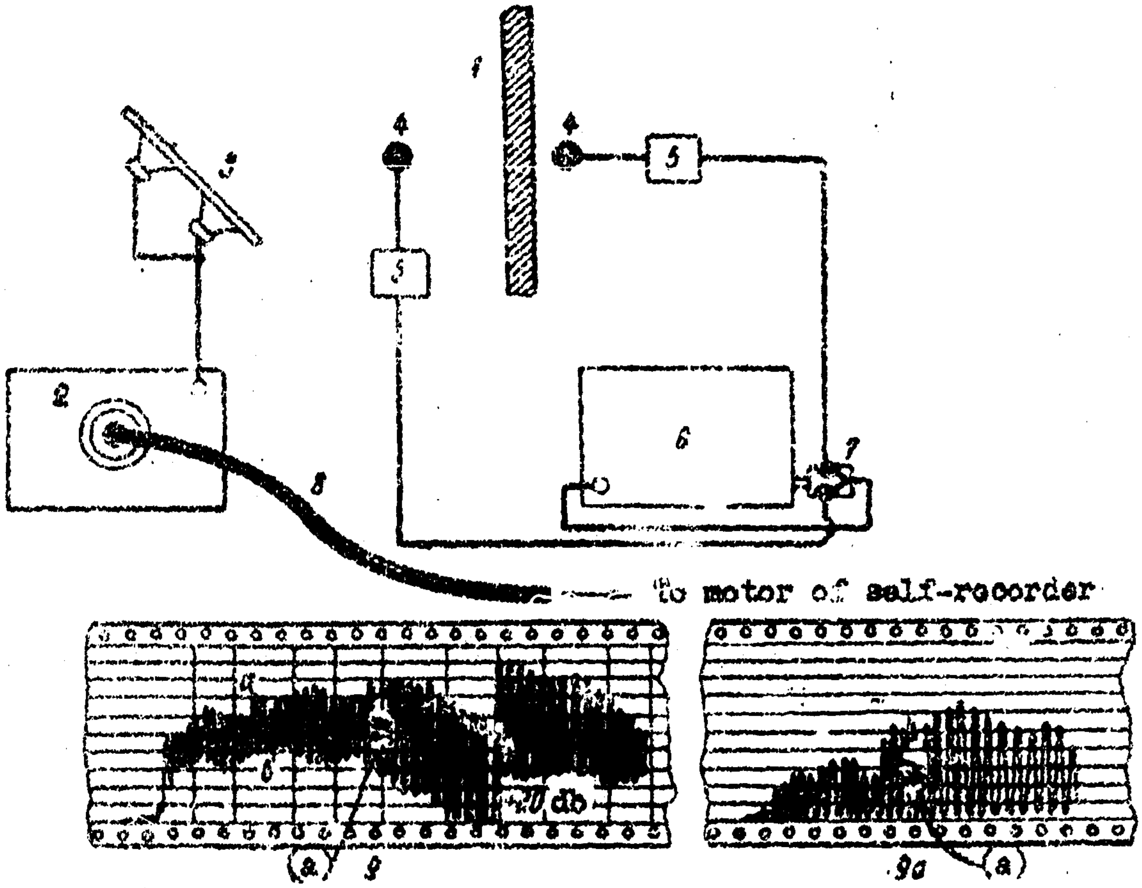


Figure 105. Automatic system of measuring sound isolation with the aid of a level recorder.
1 - The structure under test; 2 - Sound generator; 3 - Loudspeakers; 4 - Microphones; 5 - Amplifiers; 6 - Pen-recorder; 7 - Changeover switch for the shaft of the recorder motor; 8 - Flexible shaft; 9 - Sample of recording on the tape of the pen-recorder; 9a - Same, with automatic regulation of the sound level.
(a) Value of the sound isolation.

Due to nonuniformity of the frequency characteristics of the loudspeakers, the traces may meander considerably, which complicates determination of the value of the sound isolation at the various frequencies. Introduction of a level regulator into the system enables a

record to be obtained for any fixed frequency band. In this record, the values of sound isolation are given directly by the ordinate of the upper part of the trace (recording 9a, and in Figure 105). The level may be controlled either by a second recorder, or with the aid of a compression circuit in the generator (if available).

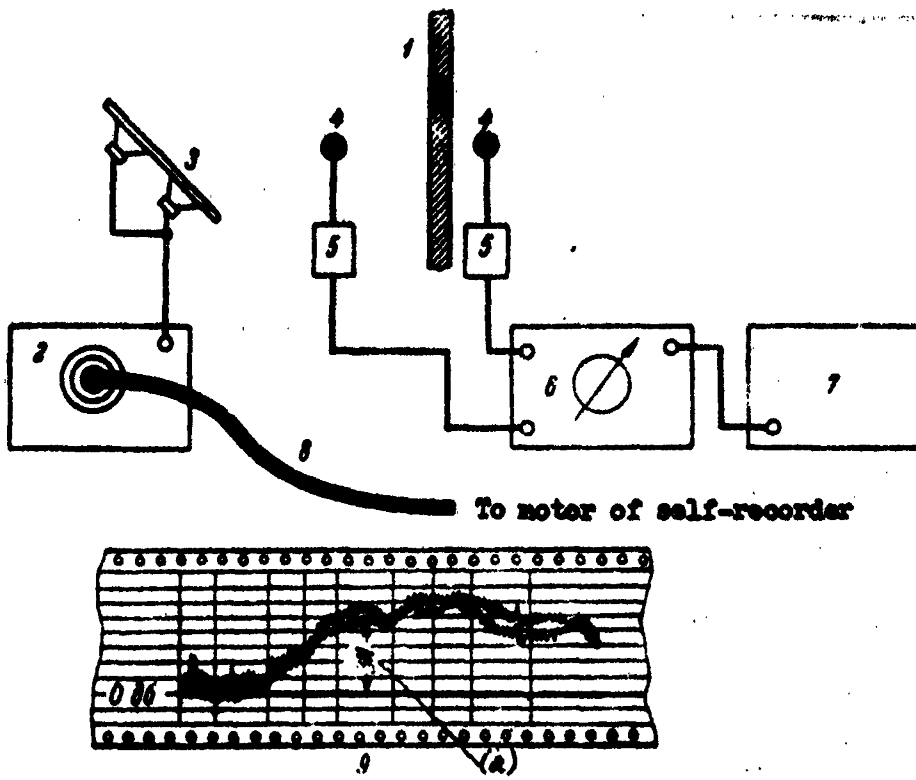


Figure 106. System for measuring sound isolation with the aid of a logarithmic logometer and level recorder.
1 - The structure under test; 2 - Sound generator; 3 - Loudspeakers; 4 - Microphones; 5 - Amplifiers; 6 - Logarithmic logometer; 7 - Level recorder; 8 - Flexible shaft; 9 - Sample of record on tape of the pen recorder.
(a) Value of the sound isolation.

Figure 106 shows a system proposed by the present author for measuring sound isolation with the aid of a special instrument, the logarithmic logometer [According to a Russian technical dictionary, a "logometer" is a device for measuring the ratio of electrical currents — Translator's note] (50), which gives the difference in level of two signals, in decibels. Because the frequency characteristics of

the microphones may have interacting reciprocal nonuniformities, two measurements are made, with the microphones at different locations for each measurement. This procedure also is recommended for other methods of measurement of sound isolation. The true curve of sound isolation versus frequency (although not corrected for reverberation) lies between the two curves thus obtained.

In addition to obtaining the frequency curves of sound isolation, the average value of sound isolation of structures may be evaluated within that frequency band for which the noise generators and noise meters are being used. We must warn against the practice, which occasionally is attempted, of using noisy machinery as a source of sound for this purpose. It is not difficult to show that, because the noise spectrum of machines falls off with frequency, very large errors, 15 to 20 db and higher, may occur in the value of the average sound isolation for a wide band of frequencies.

CHAPTER 9

LOCAL AND PERSONAL MEASURES OF PROTECTION AGAINST NOISE

34. Sound Isolating Housings for Machinery and Other Equipment

The noisiest machinery and equipment are covered with sound isolating housings. With the aid of housings, noise is reduced not only in adjacent compartments, but also in the compartment of the noise source. The weight of sound isolating housings is less than the weight of sound isolating bulkheads because the walls of the housing are directly adjacent to the source of noise.

Housings are made of dense materials such as wood, metal and plastic compositions. The inner surfaces of the housing are coated with a layer of sound absorbing material (Figure 107). To avoid overheating of the machine, the housing is provided with a ventilation system, the walls of which also are coated with sound absorbing material.

As indicated previously, for the case in which sound penetrates into a room with sound absorption, the actual sound isolation of the wall of that room is equal to (formula (139):

$$ZI_F = ZI + 10 \log \alpha \text{ db},$$

where ZI is the sound isolation of the walls when placed in open space;

α is the coefficient of sound absorption of the walls of the room (area over which the sound absorber is distributed, taken as equal to the area through which sound penetrates into the room).

In a sound isolating housing, the sound absorber is located not in the compartment which is being isolated, but in the compartment of the noise source, i.e. the machine being enclosed by the housing. It may be readily shown that in this case the actual sound isolation satisfies the condition of equation (139). As proof, we turn to equation (123), which relates the magnitude of the intensity of sound in a room with the sound absorption to the acoustic power of the

source. Using the designations from Figure 107, we may write, for moderate values of sound absorption, that:

$$J_{in} \cong \frac{4W}{\Sigma \alpha_i s_i} = \frac{4W}{\alpha s_p}, \tag{145}$$

where s_p is the area of the absorber which is on the inside surface of the housing.

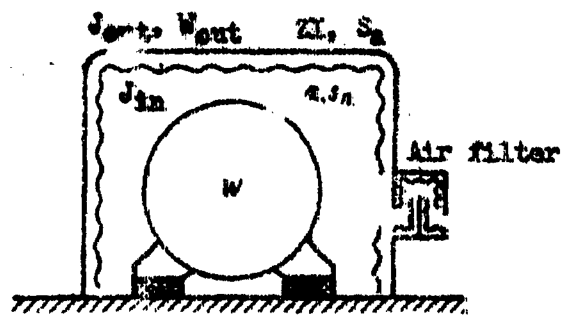


Figure 107. Symbols used in analysis of a sound isolating housing.

The intensity of sound penetrating through the wall of the housing is:

$$J_{out} = \frac{1}{4} J_{in} \cdot 10^{-0.1 ZI}$$

where ZI is the sound isolation of the wall.

The total sound power radiated by the housing into the external medium is:

$$W_{out} = S_a J_{out} = S_a \frac{W}{\alpha s_p} \cdot 10^{-0.1 ZI} .$$

It is apparent that the actual sound isolation of the housing may be expressed by means of the ratio of the energy of the noise source to the energy of the noise radiated by the walls of the housing. The value of the sound isolation of the housing, in decibels, is equal to:

$$ZI_H = 10 \log \frac{W}{W_{out}} = ZI + 10 \log \left(\alpha \frac{s_p}{S_a} \right) . \tag{146}$$

If the absorber is applied to the entire internal surface of the housing, i.e. $s_p = S_a$, then:

$$ZI_F = ZI + 10 \log \alpha \text{ db}, \tag{147}$$

which is the same as expression (139). As before, the actual sound isolation of the walls is greater the more effectively the sound absorber is applied to them. In the absence of a sound absorber in the housing, its sound isolation drops sharply, approaching zero.

For an expression of the sound isolation of the housing walls as a function of their thickness and the sound frequency we may use expression (104):

$$ZI = 20 \log (fG) - 60 \text{ db}, \tag{148}$$

where G is the weight per m^2 of the housing wall in kg.

The chart of Figure 108, constructed according to the formulas presented in the foregoing, enables rapid evaluation of the sound isolation of housings with single walls, of steel and duralumin, at various frequencies, taking into account the values of the coefficient of absorption of the internal coating of the walls at these frequencies. The inflexion points in the upper portions of the curves of sound isolation represent the loss of sound isolation which occurs at the frequencies of resonance of coincidence (formulas 76 and 77). In computing the actual sound isolation of a housing taking into account the ventilation openings, the value of ZI_F obtained must be corrected by the factor shown in Figure 96.

Example 38. The problem is to determine the approximate acoustic effect of a sound isolating housing of steel 2 mm thick, as a function of frequency. The wall of the housing is coated on the inside with felt 12.5 mm thick. The housing is equipped with two air filters, whose total area is 10 percent of the area of the walls of the housing, and whose sound isolation ZI_{vent} is a function of the frequency as follows:

f, cycles................	125	250	500	1,000	2,000	4,000
ZI_{vent}....................	2	4	8	10	12	9

It is assumed that the machine within the housing is mounted on efficient sound isolating vibration mounts and there is no reason to expect deterioration of the total sound isolation of the housing as a result of transmission of sonic vibration through the foundation.

Solution. From Table 19 we copy down the coefficients of sound absorption of felt of the given thickness, rounding them off to the

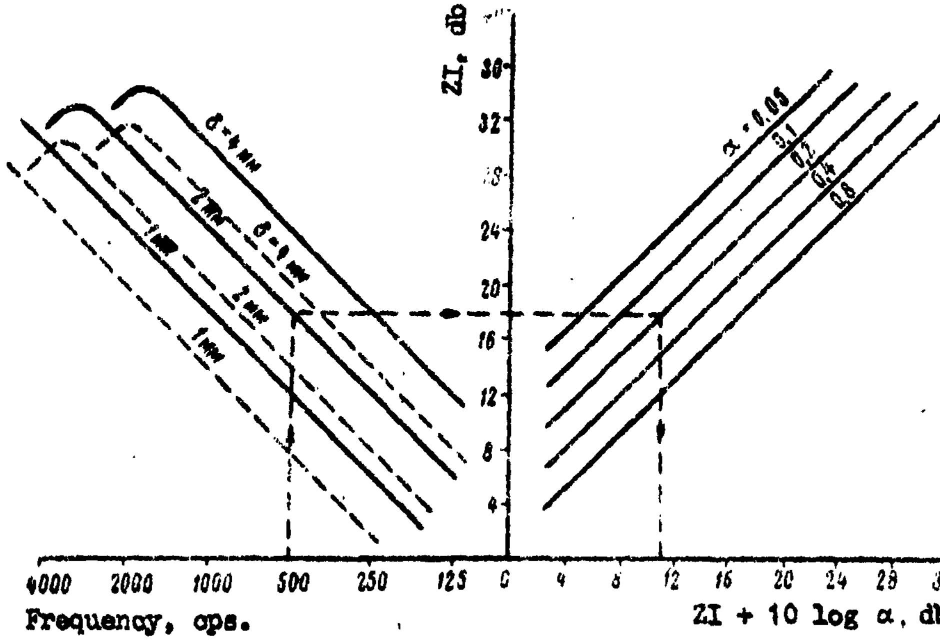


Figure 108. Chart for computing the sound isolation of a housing as a function of frequency, wall thickness δ and coefficient of absorption of the internal coating α.
——— Steel; - - - Duralumin.

first digit after the decimal. We determine the actual sound isolation of the housing wall at various frequencies according to Figure 108 (an example of finding the sound isolation for a frequency of 500 cycles is given in the graph). Next, we introduce the correction to the sound isolation value according to the curve of Figure 96, due to the influence of the filters in the housing. The computations are indicated in the following table.

f, cycles	125	250	500	1000	2000	4000
α	0,05	~0,1	~0,2	~0,5	~0,5	~0,5
ZI + 10 log α	< 2	2	11	21	27	< 28
ZI_{vent}	2	4	8	10	12	9
ZI_{total}	< 2	2	11	18	20,5	< 18

Thus it is evident that at frequencies, at least up to 250 cycles, the housing practically does not isolate the sound, due to the low value of sound isolation of felt at those frequencies, and to the relatively low thickness of the housing wall.

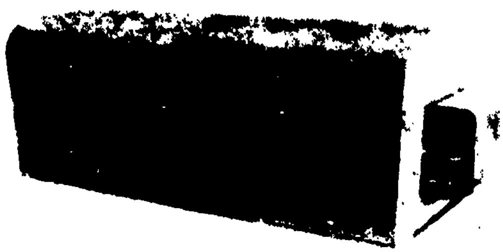


Figure 109. A sound isolating housing for an electric transformer.

The designs of sound isolating housings are extremely varied. Figure 109 shows a housing for a high rpm [high frequency] electric transformer of 3 kw power (46). The walls of the housing are of 1.5 mm steel; the inside of the housing wall is coated with a 10-mm layer of felt. The sound isolating filter, one of the side walls of which has been removed, may be seen in the illustration.

Power cables enter through the housing via hermetically sealed fittings called sound isolating "gaskets." This device consists of a cup-shaped housing containing a rubber packing collar, secured by a screw cover (Figure 111).

The frequency function of the sound insulation of this housing is very similar in character to that computed in Example 38 (Figure 110). At high frequencies sound isolation drops markedly, apparently

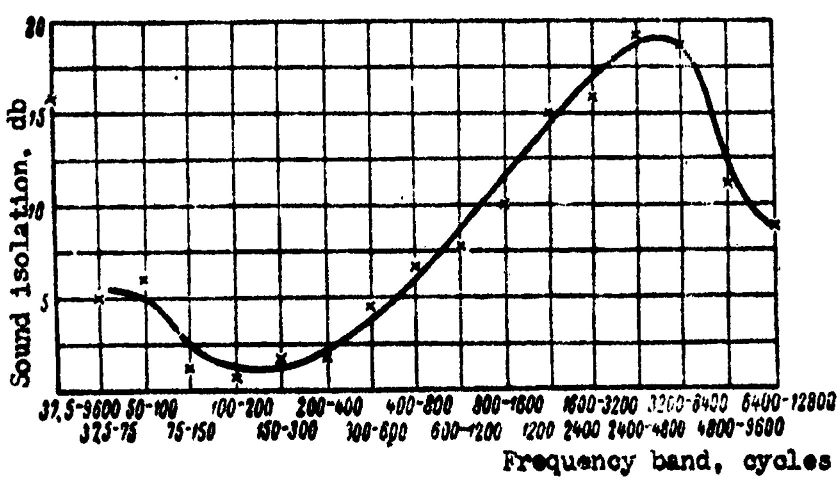


Figure 110. Frequency function of the sound isolation of the housing shown in Figure 109.

as a result of the resonance of coincidence. The most intense components in the noise spectrum of the transformer are located in the range of frequencies from 1,000 to 4,000 cycles. The sound isolation of the housing is fairly high in this range, and because of this the average value of sound isolation for the entire measured range of frequencies attains 16 db ("X" mark on the ordinate axis). Following installation of the housing, the noise of the transformer lost its irritating character, and moderate speech could be heard without effort.

The transformer operated 5 to 6 hours each day for three years without trouble (overheating, etc.).

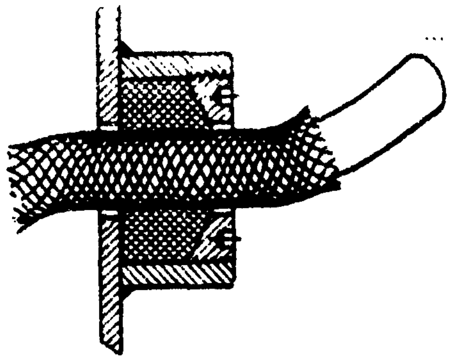


Figure 111. Cable lead-in of a sound isolating housing.

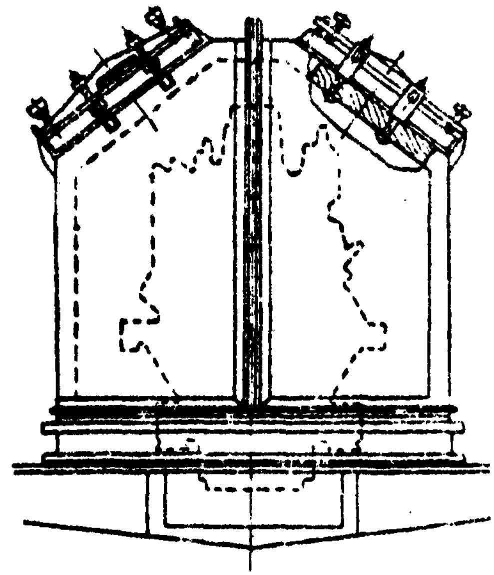


Figure 112. Sound insulating housing for the 3D6 diesel engine (150 hp at 1500 rpm; noise level 115 db).

Another example of a sound isolating structure is the housing developed for the 3D6 engine installed in the "Moskvich" class river passenger boats (48). The housing is mounted on a wooden framework, and has 1.5-mm steel walls (Figure 112). The wall of the housing is covered on the inside with 20-mm thick felting, faced with canvas and covered with perforated duralumin sheet. To enable access to engine parts requiring periodical inspection and maintenance, the walls of the housing are provided with removable ports. The design of the housing subsequently was lightened, without noticeable damage to the sound isolation properties of the housing (37).

Tubes and electric wires to the engine enter through the lower portion of the housing. At the point of passage through the housing, the propeller shaft is provided with a sound isolating stuffing box, differing from that shown in Figure 111 only in that the sealing element is cork, instead of rubber. Air intake under the housing is accomplished through an acoustic filter. The engine is operated by means of remote control, located on the bridge.

Test stand trials of the housing showed that at high frequencies the acoustic effect of the housing reaches 20 db. The irritating effect of the noise at these frequencies was reduced markedly, even with the port covers removed. The average value of the reduction of the noise of the diesel engine with covered ports was 10 to 12 db. The slightly lower average reduction in noise level in comparison with the housing shown in Figure 109 is explained by the low-frequency nature of the noise spectrum of the diesel.

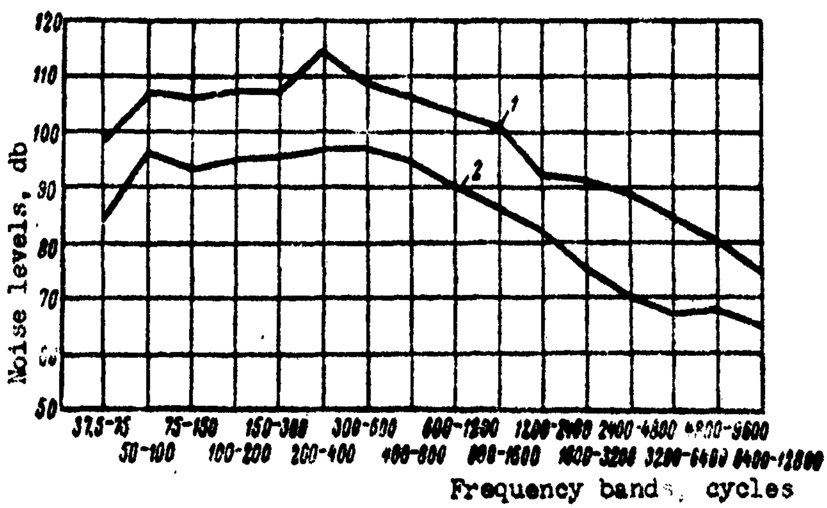


Figure 113. Reduction of the noise level of the engines in the engine room of a diesel-electric ship through the installation of a sound insulating housing.
1 - Diesel-electric ship "Kurgan" (housings not installed);
2 - Diesel-electric ship "Ivan Stepanov" (housing installed).

In the Soviet-built diesel-electric ship "Ivan Stepanov" all four diesel-generators, which are mounted on sound isolating vibration mounts, are enclosed in a common sound isolating enclosure. The walls of the enclosure consist of removable panels having a rubber seal along the perimeter, and mounted on a framework. The 4-mm thick walls of

this partition are coated on the inside with 30 mm of asbestos floss matting and calico jacketing (106).

Glass portholes are located in the housing forward and aft of each engine for observation of the operation of the diesel-generators. There are about 10 tightly fitting doors in the entire enclosure, through which the ship's personnel have access to the engines (may be opened from both the outside and inside). Toggles are passed through the housing for stopping the engines in case of emergency.

Figure 113 shows the frequency curve of noise in the engine room of the diesel-electric ship "Ivan Stepanov," in which sound isolating housings are installed. Comparison of this curve with the same curve for a ship of the same class, the "Kurgan," in which a housing had not been installed, enables evaluation of the effect of the application of this measure. It is apparent that the noise level in the sonic frequency range is reduced 15 or 20 db.

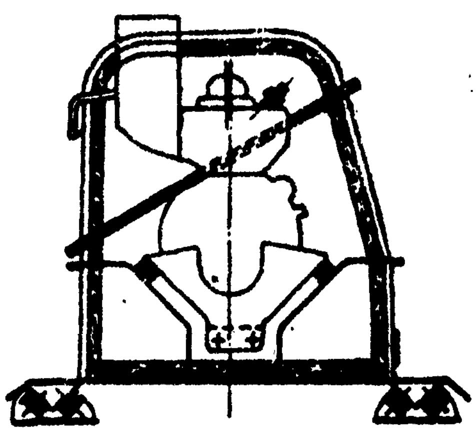


Figure 114. Sound-isolating housing with mounting of the machine on vibration isolation mounts arranged in series.

The application of a sound isolating housing-partition on the diesel-electric ship led to reduction of noise in many rooms located near the engine room. Thus the noise level at the companion way to the engine room was reduced by 13 db, and was reduced 14 db in one of the nearby cabins. The acoustic effect of the housing is practically absent in the hold areas. This is explained by the fact that the engines are uncovered when looked at from the foundation side.

Sound-protective housings are used extensively in foreign ships. Figure 114 shows a housing which has been developed, in which special vibration isolation measures have been taken. A vibration-absorbing device is located inside the housing together with the supporting structure of the engine. The housing also is mounted on vibration mount

On ships of small displacement, individual parts of the sound-protective housings may be combined structurally with parts of the bulkhead structure of the engine room. In this case the sound protective structure no longer is local, but general in nature.

35. Sound-Shielding Booths, Light Partitions, Screens, and Local Sound Absorbers

The use of sound-shielded booths and light partitions in noisy engine rooms for the confort of the engine room personnel is well known in shipbuilding practice. The booths are made of steel or wood, are mounted on sound isolating material, and are coated inside with a sound absorbing material. Devices are provided for remote control of the motors from the booths.

The designing of booths of this type differs in no way from that of sound isolating enclosures and housings (Example 38). However, installation of sound isolating housings is preferable to the installation of booths, because the former enables a reduction of noise in many compartments of the ship.

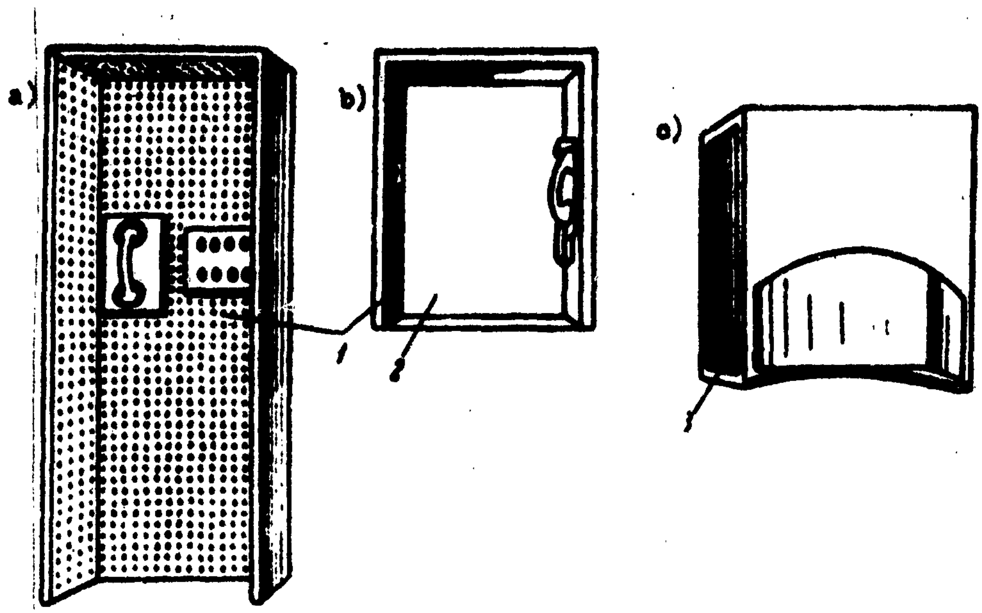


Figure 115. The arrangement of semi-enclosed telephone booths. 1 - Sound absorber; 2 - An outer transparent wall; 3 - A transparent side wall.

In rooms with not too high a noise level, telephone booths or shields simple in design and often open to enable quick access to the telephone may be useful. The user may enter the semi-enclosed booth (Figure 115-a), "dip" into the telephone shield (Figure 115-b), or push through the crown of the oval opening of the shield (Figure 115-c). Experience in the use of semi-enclosed telephone booths on Russian

ships indicates that, with the aid of these structures, the noise level at the telephone site is reduced by 10 or 12 db at medium and high sound frequencies. This greatly increases the intelligibility of telephone conversations.

Under conditions approximating the conditions of open space, on the upper decks and in very large rooms, noise may be reduced greatly at relatively short distance from the source with the aid of very simple structures, screens and compartments open on top.

Example 39. The opening of an engine room ventilation trunk is located on a superstructure deck. The trunk is provided with sound absorbers, but the noise level on deck still is very high. At a distance of 2 meters from the trunk is a partition, dividing the promenade deck. What height should the partition have to ensure that the noise level at a distance D = 10 m from the trunk opening will be 15 db lower than at the trunk, for frequencies above 500 cycles?

Solution. We turn to formula (81) and Figure 26. Because in the given case $D \gg R$, and probably $R > H$, the second quantity in parentheses in formula (81) is approximately zero. Expanding the first parentheses according to Newton's binomial gives:

$$N \approx \frac{H^2}{\lambda R}. \qquad (149)$$

From Figure 26 it is seen that to lessen the noise by 15 db the value N must be not less than 3. The required height of the partition is equal to:

$$H = \sqrt{\lambda R N} = \sqrt{\frac{340}{500} \cdot 2 \cdot 3} \cong 2 \text{ м.}$$

Let us determine the lessening of noise by the partition at still higher frequencies, where the irritating effect of the noise is very great. We find the amount by which the noise is lessened from Figure 26 based on values of N (for $H \cong R = 2$ m) computed according to equation (149).

If the partition could be placed closer to the noise source, then at the same, and even with less height, it could produce a marked lessening of noise, even for frequencies below 500 cycles. In this case the computation is performed according to the precise formula (81).

f, cycles	1000	3000
N	6	18
Lessening, db	17	~22

Occasionally combinations of separate sound absorbing elements are used for reducing noise. Figure 116 (left) shows "volumetric" sound absorbers produced by a French company. The length of the sound absorbers is approximately 0.5 m; they are suspended in groups on the overhead or walls in the vicinity of the noise source. These elements are protected on the outside by perforated metal sheets.

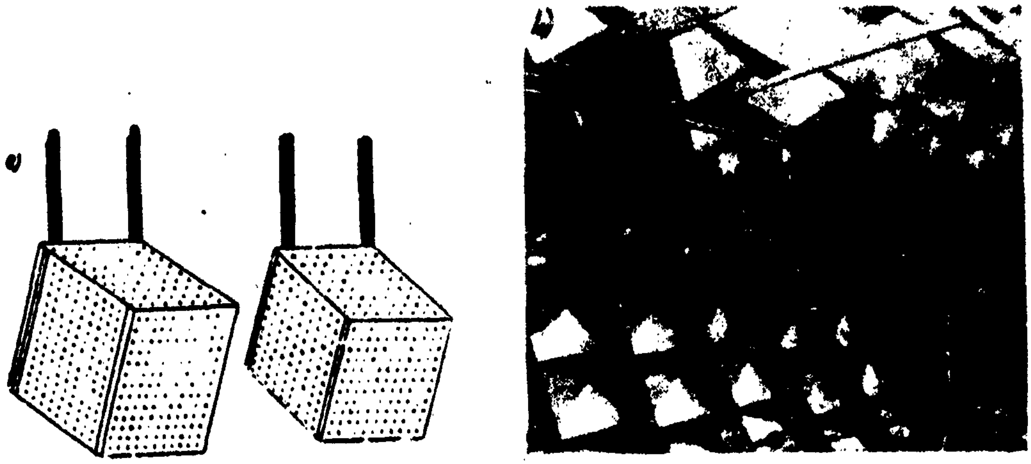


Figure 116. Two designs of absorbers and the manner of their attachment. Left - local volumetric absorbers; right - absorbers (pyramids of Meyrli) as installed in the engine room of a ship.

Due to their relatively large dimensions, the absorbers have considerable sound absorption characteristics, beginning at very low frequencies. Thus an absorber 0.5 to 0.6 meters in length ensures a sound absorption of 5 or 6 to about 10 or 12 units at a distance of 1.2 to 1.5 meters. The first figures relate to frequencies of 250 - 500 cycles, and the second to frequencies of 2 to 4 kc.

The absorbers may have the shape of wide pyramids, covered by perforated sheets (pyramids of Meyrli [Meyrli - phonetic spelling; a non-Russian name]). These pyramids are utilized as local absorbers or may be secured to the entire surface of hull sides and overheads. Their width is equal to the frame spacing; the pyramids are secured to the frames and beams by means of simple devices and if necessary may be removed (Figure 116, right). With the use of these pyramids, the noise level in the engine room of a British ship was reduced by 6 db (165). For Russian inland waterways ships, easily installed volumetric screens for sound absorption, covered with perforated vinyl plastic, have been developed. The screens are fastened along the sides and frames after the sides have been painted (37).

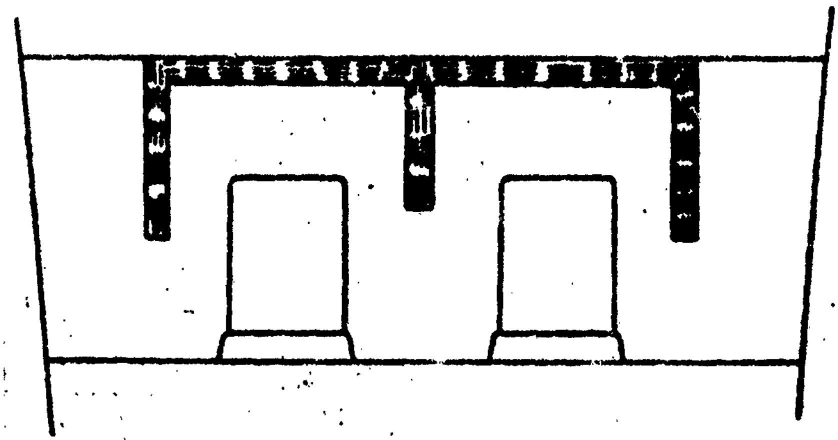


Figure 117. Diagram of the arrangement of sound absorbing screens suspended over engines.

Sound absorbing screens, suspended above and around noisy machinery also are used (Figure 117). The design of the sound absorbing screen is determined by the character of the noise. With a high-frequency noise spectrum, it is advantageous to introduce solid walls in the middle of the vertical screen. The surface of the screen thus, in effect, is doubled, without changing its thickness. If the low frequency components must be lessened in the vicinity of an engine, a maximum thickness of the absorber must be provided. In this case the introduction of solid walls within the shields, reducing the thickness of the absorber by one-half, is not rational.

36. Personal Means of Protection Against Sound

The personal means of sound protection include (Figure 118): ear plugs 1, placed in the auditory canals, or covering their openings; anti-noise ear muffs 2, covering the outer ear; sound protection helmets 3.

Figure 118. Personal means of sound protection.

The simplest anti-noise device of the first type is a cylinder of plastiline and cotton, inserted in the openings of the auditory canal (ear plug of P. P. Kudryavtsev, Figure 119-a). A set of these ear plugs in a small box may be handed to personnel who periodically are located in very noisy areas. The ear plug is discarded after use.

During World War I Russian artillery personnel used an ear plug consisting of a metallic sphere with a small wing. When in

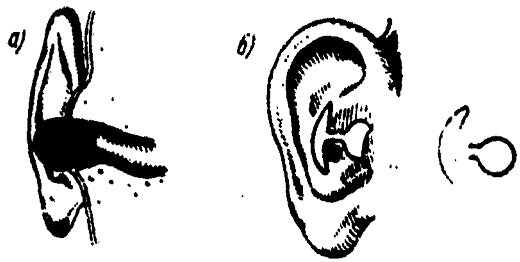


Figure 119. Very simple ear plugs.

use, the wing rested against the internal surface of the hollow of the external ear, and the sphere, having relatively large mass, completely blocked off the entrance to the auditory canal without irritating it (Figure 119-b). This ear plug also may be made of solid rubber or plastic. It provides a reduction of 15 or 20 db of noise (i.e. raises the threshold of hearing) in a broad frequency range, especially at high sonic frequencies. The device in the form of ear muffs, completely covering the external ear, also reduces noise by the same, or a somewhat greater amount.

The "hermetic" ear plugs described, or ear plugs of similar design, are not very suitable under conditions in which it is desirable, in addition to lessening harmful noise, to attain more or less clear perception of speech. An ear plug of this type must comprise an acoustic band filter, admitting only that band of frequencies which ensures the intelligibility of speech. Ear plugs of this type have not been produced yet. Ear plugs of the low frequency acoustic filter type, described by I. M. Polkovskiy (85) to a certain extent meet this condition. However, that author did not meet another requirement, that of reducing the traumatic effect of noise. Statistics show that persons subjected continuously to the effect of noise lose first of all their sensitivity to high sound frequencies. Because of this it is most important to protect the ear against frequencies above 1 kc, which may be achieved with the use of a low-frequency filter; at the same time the irritating effect of the noise is reduced, and the perception of speech on a background of noise is improved slightly.

Ear protection devices designed on the principle of the low frequency filter may be designed as an ear plug (Figure 120-a), or ear muff (Figure 120-b). The elements of these devices consists of internal air spaces, and thin tubes or canals. The inertial resistance of the volume of air in the tubes and canals increases with frequency, as a result of which high frequency sounds are inhibited by the device. I. M. Polkovskiy has presented formulas enabling computation of the general dimensions of the elements of these ear devices for a given value of the excluded frequency.

Of the personal means of noise protection the greatest value of noise reduction (up to 25 or 30 db) is afforded by sound isolating helmets. At such high values of sound elimination, the conduction of

Figure 120. Anti-noise devices of the low frequency acoustic filter type.

sound by bones begins to play a role. Thus the helmet must cover not only the ears and head, but also the cheek bones, the forehead and back portions of the jaw bones. Such a helmet would be inconvenient to wear.

Simultaneous application of helmets and ear plugs may increase the effect of the sound protection devices by an additional slight amount. The total value of absorption at frequencies above 1 kc in the case of a helmet covering only the ears and head is 30 or 35 db, but in the case of a helmet also covering part of the face, is up to 40 db. This apparently is the limiting value of noise reduction which may be attained by means of the use of personal means of protection against noise. The limiting value of the reduction apparently is due to the conductivity of sound by the bones.

37. Active (Interference) Means of Noise Reduction

During recent years reports have appeared on the local control of noise by the interference method. The essence of this method is explained in Figure 121. In a compartment in which it is desirable to lessen the noise is a microphone, the signal from which is amplified and transmitted in counterphase into the same room. In this way zones of reduced noise are formed. By means of the selection of the location of the radiating loudspeaker or group of loudspeakers, the location of one of these zones at a desired point of the room, such as the region of the head of any permanent machine operator may be accomplished. At the same time, a more or less marked strengthening of noise may be noted at other points in the room. For the purpose of reducing the number of such points and for more uniform "phasing" of sound, the back of the loudspeaker is enclosed in a hermetically sealed box containing a sound absorbing material.

Foreign aviation experience shows that with this method noise at individual points of aircraft cabins may be lessened by 10 to 12 db. A practically noticeable acoustic effect appears only at low frequencies,

or 200 to 300 cycles, where the wave length, and consequently the dimensions of the interference zones are sufficiently large.

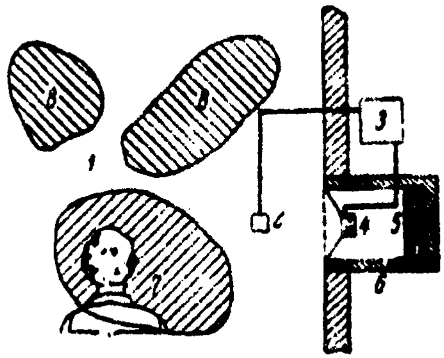


Figure 121. The interference method of local reduction of noise. 1 - Noisy compartment; 2 - Microphone; 3 - Amplifier and phase changer; 4 - Loudspeaker; 5 - Sound absorber; 6 - Sound-isolated box; 7 - Zone of local reduction of sound; 8 - Zones of local intensification of sound.

Application of this method in industry (121) also is known, in which the noise of a powerful transformer was reduced in the necessary direction by 12 to 15 db. For this purpose three interference circuits were used, each of which was tuned to its own frequency.

The interference method of sound absorption has, in essence, not yet evolved from the stage of experimental development, and requires an investigation to be conducted at each site where it is intended to be applied. Despite all the disadvantages of this method, it deserves attention as one of the few methods for controlling low frequency noises, which are usually poorly isolated by structural sound protective arrangements.

CHAPTER 10

THE SILENCING OF NOISE PROPAGATING THROUGH

VENTILATION DUCTS

38. The Physical Bases of the Silencing or Muffling of Sound in Ventilation Ducts

Shipboard ventilation systems are sources of intense noise. The very powerful ventilators of engine rooms (and the boiler rooms of steam-propelled vessels) radiate especially strong noise. However, the noise of the general ships' ventilation system also may be rather unpleasant.

With the aid of special measures (cf. Chapter 19), these noises may be reduced to a certain extent in the ventilator itself. In most cases the reduction of noise of ventilators at the source is inadequate and this requires that the noise be reduced in the course of its propagation through the ventilation ducts or at their openings.

The problem of absorption of noise in ventilation ducts has been dealt with at length in the literature. Prior to World War II the basic works of A. I. Belov (20), and the works of the German scientists V. Pining and V. Zeller were published. After World War II many investigations were performed in the USSR by Ye. Ya. Yudin (114, 115) and I. K. Razumov (87). The above-mentioned works present not only the theoretical bases of sound muffling in air ducts, but also describe the corresponding structure of mufflers for industrial buildings, apartment houses and test stands. However, mufflers and silencers developed for other purposes may not in all cases be applied to shipboard ventilation systems. Therefore, investigations dealing with mufflers especially designed for shipboard ventilators are of interest.

Active silencers, utilizing a layer of sound absorbing material, are used almost exclusively for muffling noise in ventilation ducts. All of the silencers which are used may be divided into two basic types: the channeled (kanaloviy) and the chambered (kamerniy), (Figure 122) on the basis of their design and principle of operation. In silencers of the first type, the absorption of sound is accomplished

during its propagation through channels of various shapes and dimensions, coated with a sound absorber. In silencers of the second type, in addition to the effect of sound absorption by sound absorber layers, the effect of an expansion chamber also is employed. That effect consists of the reflection of sound at the entrance and exit of the chamber, and of the reduction of the density of the sound energy as a result of its expansion in the volume of the chamber. Thus, ventilation system silencers of the second type utilize, in addition to active elements, reactive elements, i.e. elements not connected directly with energy loss, but causing a reduction of sound in remote portions of the duct.

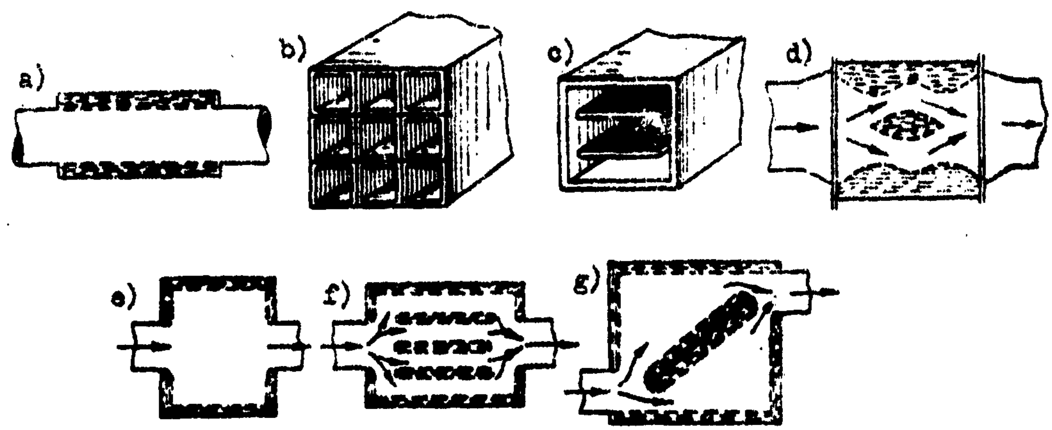


Figure 122. Types and arrangements of silencers and mufflers for reducing ventilation system noise.
Channel type absorbers: (a) Tubular; (b) Honeycomb (cellular); (c) Lamellar; (d) Absorber with curvilinear channels.
Chamber type absorbers: (e) Single-chamber muffler; (f) Chambered lamellar muffler; (g) Chamber muffler with screens.

The simplest of the silencers of the first type is the tubular muffler, or muffler with a sound-absorber tube, consisting of a layer of sound absorbing material under a netting or perforated sheet, located in the duct. As indicated by A. I. Belov, the reduction of noise in this muffler is proportional to its length l, the perimeter of the cross-section P, and is inversely proportional to the cross-sectional area of the duct S:

$$\Delta\beta_{\kappa} = 1,1\alpha' \frac{P}{S} l \text{ db}; \qquad (150)$$

where α' is the coefficient characterizing the absorption of sound by the layer (for greater detail, cf. Paragraph 39). The values P, l and S are expressed in meters.

In the case of a square duct of width D, or a round duct of diameter D the reduction is equal to:

$$\Delta\beta_{\kappa} = 4,4\alpha' \frac{l}{D} \text{ db.} \tag{151}$$

This formula indicates that to increase the reduction of sound in the silencing channel, its transverse dimension must be reduced. The concept of reducing the transverse dimension of the sound absorbing element while retaining a cross section necessary for air supply is realized by use of a honeycomb muffler (Figure 122-b). To determine the noise reduction of a honeycomb muffler with identical cells it is sufficient to compute the reduction occurring in one cell.

The lamellar channel muffler (Figure 122-c) differs from the honeycomb type only in the form of the channels, which are formed by sound absorbing sheets. The reduction of sound in a lamellar muffler is computed according to the following formula:

$$\Delta\beta_{1am} = 2,2\alpha' \frac{l}{a} \text{ db,} \tag{152}$$

where a is the distance between the sheets.

In the experiments of I. K. Razumov he found that, contrary to formula (150), the reduction of sound in the channel was proportional to $\sqrt{\frac{P}{S}}$.

The reduction of sound in chamber-type absorbers may be computed according to the formula:

$$\Delta\beta_{cham} = 10 \log \frac{\alpha S_{in}}{S_{\kappa}} \text{ db,} \tag{153}$$

where α is the coefficient of absorption of the coating;

S_{in} is the total area of the internal surfaces coated with sound absorber in the chamber;

S_{K} is the cross-sectional area of the duct.

Supplementary sound absorbing plates or sheets (Figure 122-f) also may be placed in the chamber of the absorber. With an equal number

of plates, the lamellate chamber muffler ensures more reduction of sound than a lamellate channel muffler due to the effect of the expansion chamber. However, its aerodynamic resistance is greater than the resistance of the channel muffler.

An acoustic effect due to a change in the cross-sectional area of an air duct occurs not only in chamber type mufflers, but also at branches, constrictions and expansions of the air ducts. Formula (87), introduced in the preceding material, applies to the value of the coefficient of reflection of sound ϑ' at the site of a change in the cross-sectional of a sound channel. Using the designations of sketch b of Figure 123, that formula takes on the following form:

$$\vartheta' = \left| \frac{1 - \frac{S_1}{S}}{1 + \frac{S_1}{S}} \right| .$$

The coefficient of the transmission of sound through the place of the change in cross section is equal to (in terms of its engery):

$$\alpha = 1 - \vartheta'^2 = 1 - \left| \frac{1 - \frac{S_1}{S}}{1 + \frac{S_1}{S}} \right|^2 = \frac{4 \frac{S_1}{S}}{\left(1 + \frac{S_1}{S}\right)^2} .$$

The reduction of sound at a change in cross section (either an increase or a decrease in the section) is equal to:

$$\Delta\beta_{sect.} = 10 \log \frac{1}{\alpha} = 10 \log \frac{\left(1 + \frac{S_1}{S}\right)}{4 \frac{S_1}{S}} \text{ db}. \qquad (154)$$

This formula holds when the transverse dimensions of the duct are considerably less than the length of the sound wave.

The reduction of sound in the case of branching of the duct, i.e. a "T" joint, may be determined in similar manner. The sound level at cross section S_1 (cf. sketch a in Figure 123) is less than the level before the T-joint by the following value:

$$\Delta\beta_{T1} = 10 \log \frac{\left(1 + \frac{S_1}{S} + \frac{S_2}{S}\right)^2}{4\frac{S_1}{S}} \text{ db.} \tag{155a}$$

The sound propagating through the cross section S_2 is diminished by the following value:

$$\Delta\beta_{T2} = 10 \log \frac{\left(1 + \frac{S_1}{S} + \frac{S_2}{S}\right)^2}{4\frac{S_2}{S}} \text{ db.} \tag{155b}$$

At $\frac{S_2}{S} = 0$, expression (155a) transforms into formula (154). The values $\Delta\beta_{T1}$ and $\Delta\beta_{T2}$ are plotted on the graph shown in Figure 123 as a function of the ratios $\frac{S_1}{S}$ and $\frac{S_2}{S}$; the lower curve corresponds to a change in the cross section of the duct, and the other curves correspond to T-joints.

Example 40. The cross-sectional areas of two air ducts branching off the main air duct are 0.2 and 0.8 of the cross-sectional area of the main duct. The problem is to determine the amount of reduction of sound in the branches at frequencies for which the dimensions of the air duct cross sections are considerably less than the wave length of the sound.

Solution. According to the graph of Figure 123, we find for $\frac{S_1}{S} = 0.2$ and $\frac{S_2}{S} = 0.6$:

$$\Delta\beta_{T1} = 7 \text{ db.}$$

(cf. dashed line on graph (------)).

Similarly, for the second branch of the air duct (taking the values of the ratios of areas shown in circles on the graph):

$$\Delta\beta_{T2} = 1 \text{ db.}$$

This computation is noted on the graph by a broken (-·-·-) line.

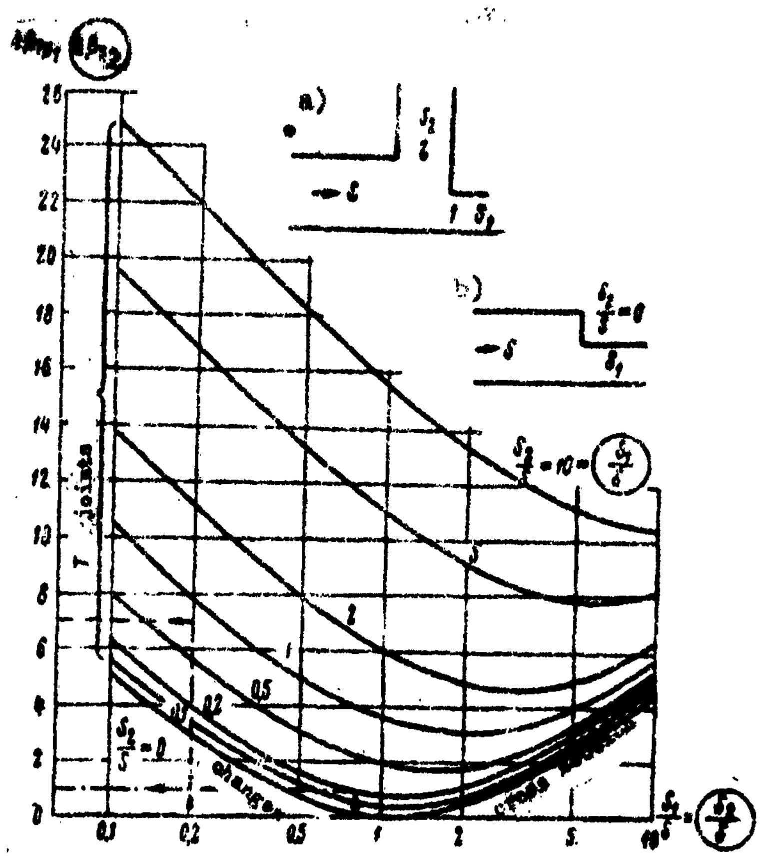


Figure 123. The reduction of sound in T-joints and at changes in the cross section of the duct.

It may be noted that when, as in the present example $\frac{S_1 + S_2}{S} = 1$ (and this is desirable from the point of view of aerodynamic considerations), expression (155) assumes the following simple form:

$$\Delta\beta_{T1} = 10 \log \frac{S}{S_1}; \qquad (156a)$$

$$\Delta\beta_{T2} = 10 \log \frac{S}{S_2}. \qquad (156b)$$

A reduction of sound in a duct is observed not only when the cross-sectional area of the duct is changed, but also when it makes sharp turns. At medium sound frequencies this diminution may be evaluated by $\Delta\beta_{turn} = 2 - 3$ db.

The reduction of sound at the outlet grille of air ducts is $\Delta\beta_{gr} = 3 - 5$ db.

In some cases mufflers are attached to the ends of air ducts. Certain types of these end-, or as they still are called, screen absorbers are shown in Figure 124. Their absorbing effect varies from 5 to 15 db.

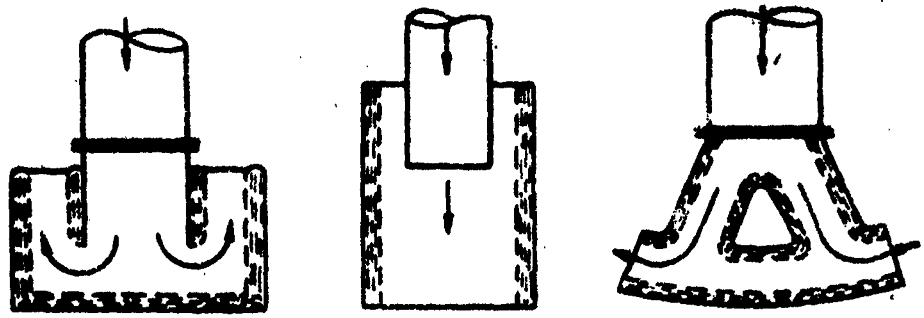


Figure 124. Mufflers mounted at the ends of air ducts.

39. Design and Analysis of Shipboard Ventilation System Mufflers

Because under shipboard conditions loss of pressure in the ventilation system is very undesirable, muffler designs must be selected which have minimal aerodynamic resistance. Table 19 gives the values of this resistance for the various designs of mufflers which are shown in Figure 122 (69).

A muffler of the sound absorbing tube type has the least aerodynamic resistance. At an air flow speed of 20 m/sec it does not exceed 4 mm of water per running meter of the muffler tube. This holds for cases when, as indicated in Figure 122-a, the surface of the sound absorber is fitted flush with the inner surface of the ventilation duct, i.e. the cross-section of the latter does not change at the place of installation of the muffler.

Honeycomb and lamellar channel mufflers have greater aerodynamic resistance. It is especially great in mufflers with curvilinear channels and in chamber mufflers with screens. Because of this, attention in the following paragraphs is focused on tubular and lamellar channel mufflers.

In the basic formula for the acoustic analysis of mufflers (formula 150), the coefficient α' usually is taken as a function only of the value of the coefficient of absorption of the coating. This function, expressed in the form of a table, however, has not been

substantiated by experience: at high sound frequencies the value of the reduction of noise by the muffler drops even at high values of the coefficient of absorption of the coating. The drop in acoustic effect is caused by the fact that at high frequencies the length of the sound wave becomes considerably less than the cross-sectional dimensions of the muffler and its center portion conducts sound freely.

Table 19

Aerodynamic Resistance of Ventilation System Mufflers

Type of Muffler	Speed of air current, m/sec	Aerodynamic resistance, mm water
Muffler of the sound absorbing tubing type, with cross-sectional area of $S = 0.02 - 0.1$ m^2 (Fig. 122, a)	15 - 20 (in muffler)	2 - 4 per meter
Honeycomb, or lamellate muffler, 300 mm in length, 3 cells with section 38x175 mm each (Fig. 122, b c)	17 (in air duct section 175 x 175 mm before muffler)	17
Muffler with curvilinear ducts 250 mm in length (Fig. 122, d)		~200
Chamber mufflers with dimensions 750x750x750 mm, with no sound absorbing coating		
without screen (Fig. 122, e)		30
with 1 horizontal screen		45
with 2 horizontal screens		80
(of type in Fig. 122, f)		
with 1 vertical screen		85
with 1 diagonal screen		
(of type in Fig. 122, g, but ducts in center of vertical wall of chamber)		115

The values of the coefficient α' for two sound absorbing materials which may be applied in shipboard ventilation system mufflers, VT-4 (kapron batting)[kapron is approximately the same as nylon-6. VT-4 is a designation for wadding or batting made by compressing kapron fibers—Translator's note] and asbestos floss, are shown in Table 20. The values of α' were obtained from the data of numerous measurements of sound muffling tubes of various lengths and various cross sectional shapes in ventilation ducts (69).

Example 41. The problem is to compute the frequency characteristics of the noise reduction of a sound absorbing tube (tubular absorber) 375 mm in length and 125 mm diameter, coated with 50 mm-thick VT-4 material.

Table 20

Values of the Coefficient a′ for VT-4 Material ("Kapron") and Asbestos Floss

Name of Material	Thickness, mm	Frequency Band, Cycles												
		85—120	120—170	170—240	240—340	340—480	480—680	680—960	960—1360	1360—1920	1920—2720	2720—3840	3840—5440	5410—7680
Material VT-4	25	—	0,11	0,17	0,4	0,51	0,63	1,0	1,1	1,8	1,7	1,0	0,8	0,46
	50	0,15	0,23	0,53	0,76	0,9	1.0	1,5	1,8	1,9	1,36	1,14	0,98	0,6
	75	0.23	0.29	0,63	0.92	1.20	1,4	1.9	2.0	1.7	1,7	1,14	0,98	0,63
Asbestos Floss	25	—	0,11	0,17	0,4	0,57	0,6	1.1	1.25	1,2	1,2	1,1	1,0	0,67
	50	0,15	0,23	0,28	0,45	0,57	0,51	0.78	1,14	1.7	1.9	1,5	1,1	0,67

Solution. The computation is performed according to formula (151), utilizing the data of Table 20. First of all we compute:

$$4,4\frac{l}{D}=4,4\frac{0,375}{0,125}=13.$$

The remainder of the computation is shown in the table below.

Frequency band, cycles	85—120	120—170	170—240	240—340	340—480	480—680	680—960	960—1360	1360—1920	1920—2720	2720—3840	3840—5440	5440—7680
α'	0,15	0,23	0,53	0,76	0,9	1,0	1,5	1,8	1,9	1,36	1,14	0,98	0,6
$4,4\frac{l}{D}\alpha$, db	2	3	7	10	12	13	20	23	25	18	15	13	8

The results of the computation are plotted as a curve in Figure 125. For purposes of comparison, points corresponding to the data of laboratory acoustic investigation of several mufflers with sound-absorbing coating of VT-4 material are plotted on the same graph. The geometric dimensions and shapes of the cross-sections of these mufflers varies, but the ratio $\frac{P}{S}l$ remains unchanged, which according to formula (152) must ensure the same value of noise reduction at each given frequency. The computed data are in fairly good agreement with the experimental data.

Let us consider now another design of shipboard muffler, a lamellar muffler for engine room ventilators. Some very powerful ventilators, installed on a ship, were radiating a noise the level of which exceeded 120 db at certain frequencies. The noise was at the ship's bridge, where the inlet grating of the ventilator shaft was located, and also in the engine room, where it prevented the reception of commands and was traumatic with respect to the hearing organs of the personnel.

For the purpose of reducing the noise the internal surface of the ventilation trunk was coated with a sound absorber consisting of a layer of cotton wadding, soaked in a fire-inhibiting solution. In addition, supplementary sound absorbing panel-plates were installed in the trunk. This type of muffler received the name of "trunk muffler." Figure 126 shows the manner in which the sound absorbing panels and

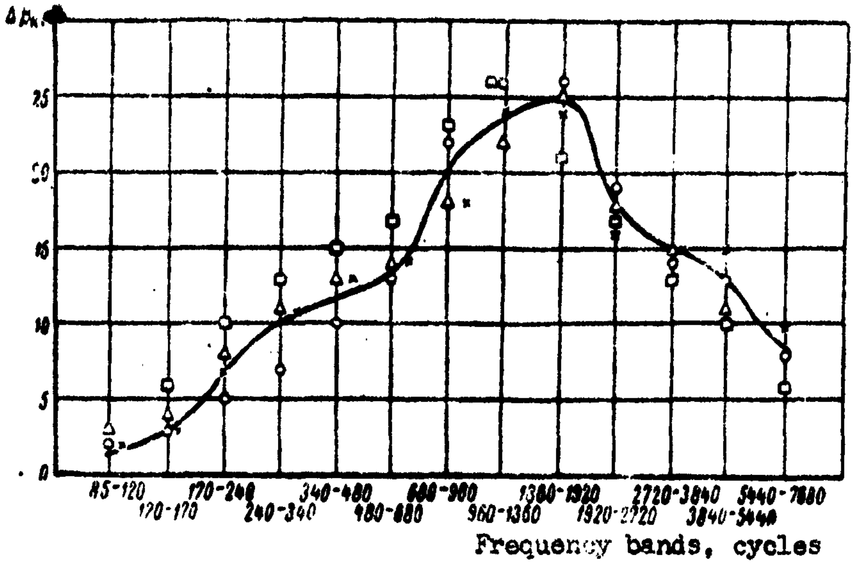


Figure 125. Frequency characteristic of the reduction of noise in a tubular ventilation system muffler of length 3 calibers (sound absorber: VT-4 material, thickness δ = 50 mm).
—— Data computed for muffler: D = 125 mm, l = 375 mm.
Other points as follows:
- o D = 125 mm; l = 375 mm
- □ D = 180 mm; l = 540 mm
- △ S = 175 x 175 mm; l = 525 mm
- x S = 175 x 400 mm; l = 730 mm

According to experimental data.

the coating should be placed (and also the ventilator) to reduce noise both on deck and in the engine room.

Following installation of the sound absorbing elements in the trunk, the noise level at medium and high frequencies dropped below the threshold of the masking of speech (Figure 127), commands could be received, and the irritating effect of the noise decreased sharply.

The noise level at low frequencies practically did not change after installation of the muffler, which may be explained by the relatively low value of sound absorption of the wadding at these frequencies. Thicker layers of absorber, up to 30 cm (139) are used on German ships for increasing the acoustic effect of ventilation system mufflers at very low frequencies.

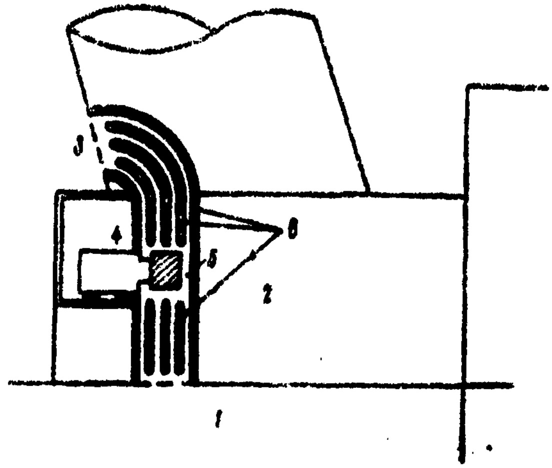


Figure 126. Placement of ventilator and absorbers in the superstructure.
1 - Ventilated engine room; 2 - superstructure; 3 - Ventilator inlet grating; 4 - Ventilator; 5 - Ventilation trunk; 6 - Sound absorbing panels and coating.

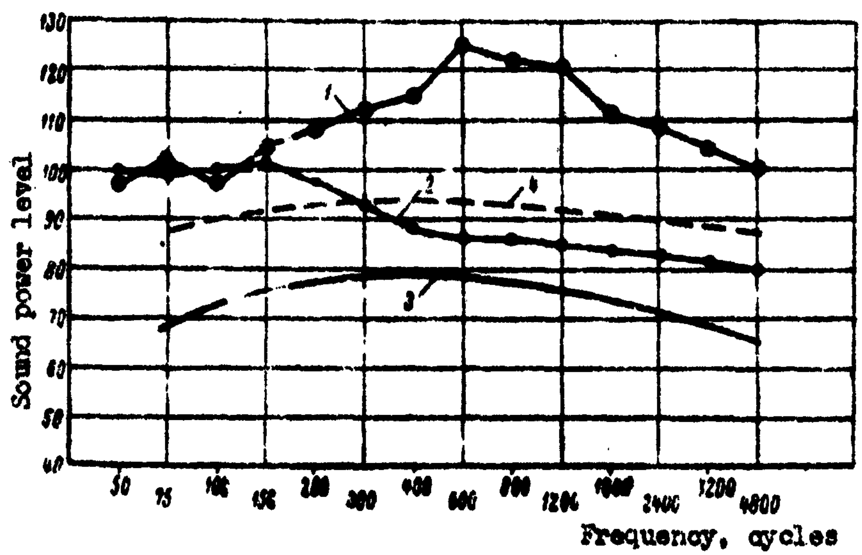


Figure 127. Noise levels in the engine room.
1 - Before installation of the mufflers; 2 - After installation of the mufflers; 3 - Speech level (loud voice); 4 - Threshold of masking of speech.

Figure 128 shows some details of trunk and channel mufflers. The detachable sound absorbing panels consist of metal or wooden frames, within which layers of quilted sound absorber are applied. The layer of the sound absorber is fixed to the walls of the absorber with mastic, glue or metal nails.

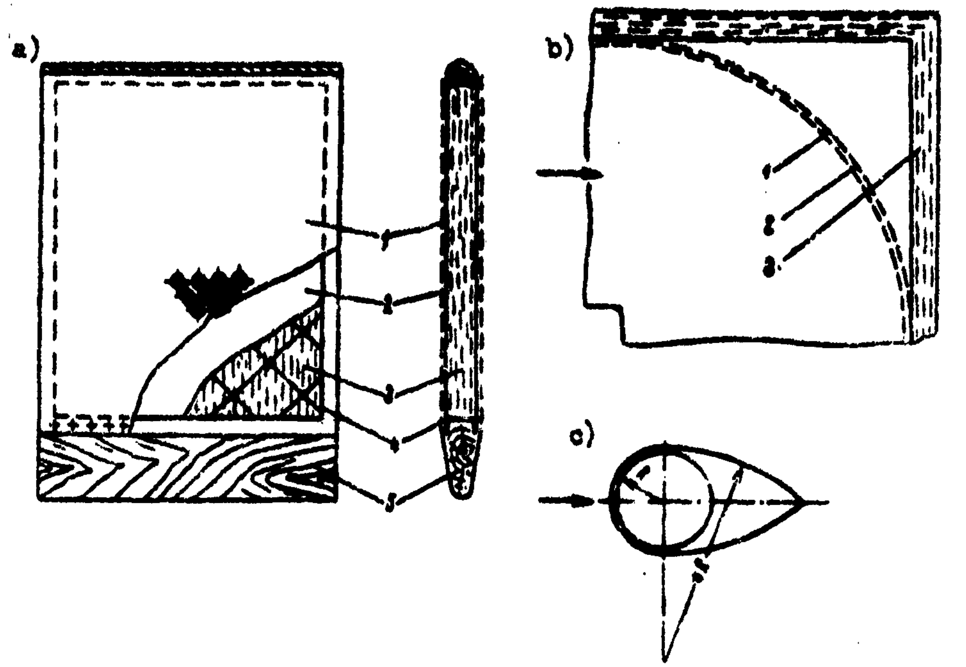


Figure 128. Details of mufflers.
(a) - Sound absorbing panel: 1 - Perforated duralumin sheet; 2 - Cloth (cheap cotton cloth, calico, moleskin); 3 - Sound absorber; 4 - Steel framework; 5 - Wood fairing; (b) - Screen at bend in trunk; 1 - Perforated metal sheet; 2 - Canvas; 3 - Sound absorber; (c) - Cross section of faired strut in air duct.

The sound absorber is covered on the outer side by a light cloth underneath a perforated metal sheet. The type of material covering the absorber is very important from the point of view of acoustics. Thus, for example, the use of a thick cloth such as canvas in place of the light cloth or perforated metal sheet may reduce the acoustic effect of the muffler at medium and high frequencies by 10 to 50 db (Figure 129).

At places where the air ducts or ventilation trunks make sharp turns perforated metal screens, backed by sound absorbing material, are placed to guide the air flow (Figure 128-b). It is useful also, to place guide vanes, which may be covered with sound absorber, at trunk or duct corners.

Various types of struts and tubes located in the air duct to reduce aerodynamic resistance and to prevent turbulence noise are provided with fairing (Figure 128-c). The same goal is served by the

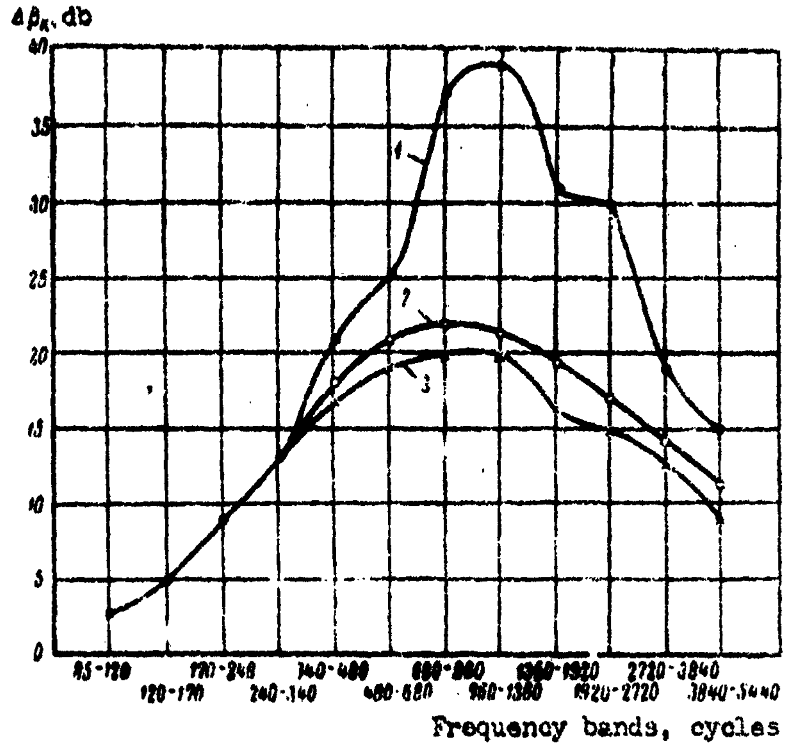


Figure 129. Effect of type of material covering the sound absorber, upon the performance of the muffler (muffler: l = 500 mm, D = 125 mm; sound absorber, VT-4 material, thickness δ = 75 mm).
1 - Sound absorber, covered with fiberglass netting or brass netting; 2 - Sound absorber covered with asbestos cloth; 3 - Sound absorber covered with glass cloth.

faired design of the trailing edge (with respect to the air movement) of sound absorbing panels, and their rounded leading edges (Figure 128-a). With proper design of sound absorbing panels their introduction into a ventilation trunk changes its aerodynamic resistance very little, because the harmful effect of increasing the surface area wetted by the air stream is compensated to a certain extent by the useful effect of regulation of the flow structure (laminarization) due to the presence of relatively slender parallel channels in the trunk.

The walls of air ducts and ventilation trunks are sufficiently resistant to excitation by the air flow. The application of double-layered walls or vibration damping coatings (Paragraph 54) also reduces

vibration of the walls and enables complete realization of the effect of mufflers installed in the air duct.

In addition to the types of mufflers described in the foregoing, "resonance" and "interference" absorbers also have been suggested. These absorbers are effective only in narrow frequency ranges, and because of this rarely are used.

In complex runs and branching of ventilation systems, the reduction of sound in all portions of the air ducting must be taken into consideration. In this, it may be determined whether mufflers are needed or not, or whether very simple absorption elements are required, on the basis of the nature of the acoustic problem and the distance between the ventilated room and the noise source.

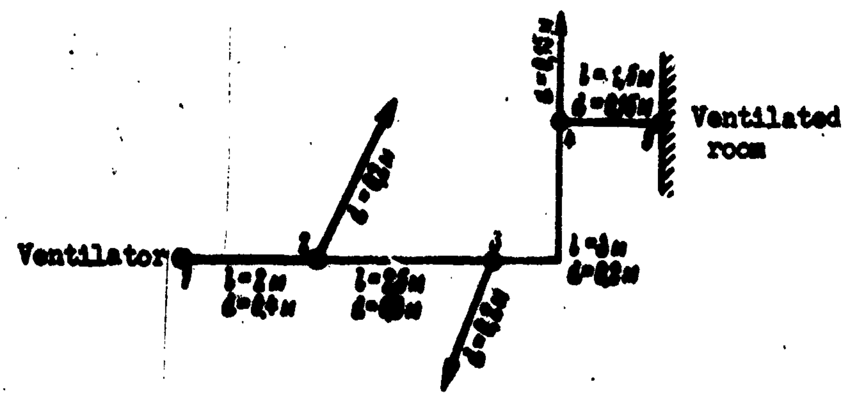


Figure 130. Computing the reduction of sound in a branching ventilation air duct.

<u>Example 42</u>. The problem is to compute the reduction of noise in the air duct depicted in Figure 130 (from point 1 to point 5). The air duct has metal walls ($\alpha' \approx 0.05$), and the frequency of sound is 400 cycles.

<u>Solution</u>. We first determine the reduction of sound in the straight portions of the duct. The computation is performed according to formula 151.

No. of section of air duct (Figure 130)	Length of section l, meters	Diameter of duct D, meter	$\Delta\beta_K = 4{,}4\alpha' \frac{l}{D}$, db
1 - 2	2	0.4	1
2 - 3	2.5	0.3	2
3 - 4	3	0.2	3
4 - 5	1.5	0.15	2
Total reduction in the straight runs			8 db

Next the diminution of sound in the T-joints, at turns in the air duct and at the outlet grating are taken into consideration. Because the wave length $\lambda = \frac{340}{400} = 0.85$ m, i.e. considerably exceeds the transverse dimensions of the air ducts, the data of Figure 123 may be utilized.

For T-joint No. 2, the ratio of the cross-sectional areas of the air ducts is:

$$\frac{S_1}{S} = \left(\frac{0{,}3}{0{,}4}\right)^2 = 0{,}56; \quad \frac{S_2}{S} = \left(\frac{0{,}2}{0{,}4}\right)^2 = 0{,}25.$$

According to the graph of Figure 123, the reduction of sound at the T-joint for these values of cross-sectional area ratio is equal to:

$$\Delta\beta_{T2} \cong 2 \text{ db.}$$

Similarly, for T-joint No. 3:

$$\frac{S_1}{S} = \frac{S_2}{S} = \left(\frac{0{,}2}{0{,}3}\right)^2 = 0{,}45;$$

$$\Delta\beta_{T3} \cong 3 \text{ db.}$$

For T-joint No. 4:

$$\frac{S_1}{S} = \frac{S_2}{S} = \left(\frac{0{,}15}{0{,}2}\right)^2 = 0{,}56;$$

$$\Delta\beta_{T4} \cong 3 \text{ db.}$$

The total reduction in the air duct (in db):

Straight runs	8
T-joint 2	2
T-joint 3	3
Turn in section 3 - 4	3
T-joint 4	3
Outlet grating	3
Total	22

40. The Calculation of Noise Penetrating Through Ventilation Ducts and Openings

Let us assume that it is necessary to determine the level of noise being transmitted from a noisy compartment I, through a ventilation air duct into a quiet compartment II (Figure 131-a). The sound conducting area of the air duct is S_K, and its sound isolation, or that of the muffler installed within it, is ZI_K.

In the first approximation, ventilation duct may be likened to a wall with a definite sound isolation value. Then, utilizing formula (136), we obtain the noise level in the isolated compartment in the following form:

$$\beta_2 = \beta_1 - ZI_K + 10 \log \left(\frac{S_K}{\alpha_{av} S_{av}} \right) \text{db}, \tag{157}$$

where $\alpha_{av} S_{av}$ is the average sound absorption in the isolated compartment.

A more precise solution requires the directionality of the radiation from the mouth of the duct to be taken into account, considered as a concentrated source of sound.

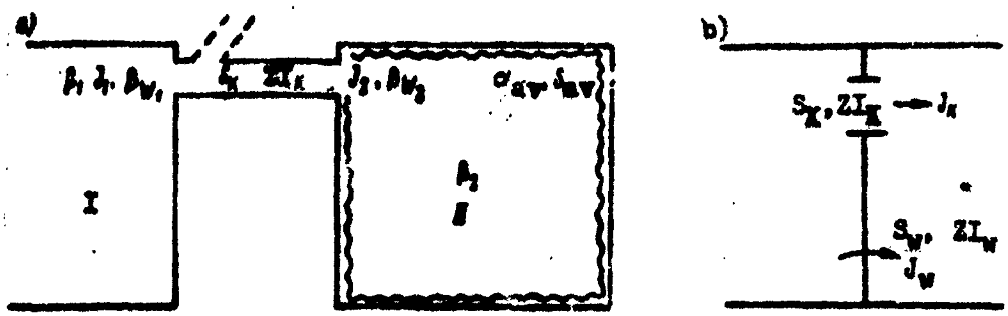


Figure 131. Computing the noise transmitted through air ducts, and the sound isolation of a muffler in the air duct.

On the basis of formula (131) we have:

$$\beta_2 = \beta_{W_2} + 10 \log \left(\frac{Q}{4\pi r_M^2} + \frac{4}{R} \right) - 40 \text{ db}, \qquad (158)$$

where β_{W_2} is the power level of sound radiated by the mouth of the duct;

Q is the directivity factor, determined as a function of the location of the mouth of the duct in the wall, according to Table 15;

r_M is the distance from the mouth of the duct, meters;

R is the room constant, equal to $R = \frac{S_{av}\alpha_{av}}{1 - \alpha_{av}}$.

Let us determine the value of β_{W_2}.

The power of the sound transmitted into the duct from compartment I is:

$$W_1 = J_1 S_K.$$

The power level of sound at the beginning of the duct:

$$\beta_{W_1} = 10 \log \frac{W_1}{10^{-16}} = \beta_1 + 10 \log S_K,$$

where β_1 is the level of the intensity of sound at the beginning of the duct.

The power level of sound at the duct exit is:

$$\beta_{W_2} = \beta_{W_1} - \Sigma I_K = \beta_1 + 10 \log S_K - \Sigma I_K .$$

Substituting the expression β_{W_2} in formula (158), we finally obtain:

$$\beta_2 = \beta_1 - \Sigma I_K + 10 \log S_K + \underline{10 \log \left(\frac{Q}{4\pi r_M^2} + \frac{4}{R} \right) - 40} \text{ db}. \qquad (159)$$

The value of the underscored portion of the formula may be determined from Figure 87 for various values of R; S_K is taken in cm^2.

Example 43. The area of a ventilation duct joining a noisy room with an isolated room is 0.04 m^2; the sound isolation of the duct (taking into account reflection of sound at the grating) is 10 db; the ventilation duct is located in the corner of the room. The problem is to determine the noise level in the isolated room which has a total wall area of 100 m^2 and an average coefficient of muffling of 0.2. The noise level in the noisy room is 100 db.

Solution. According to formula (157) the noise level in the isolated room is:

$$\beta_2 = 100 - 10 + 10 \log \frac{0{,}04}{100 \cdot 0{,}2} = 63 \text{ db.}$$

Let us determine the same level according to a more precise formula, taking into account the fact that the ventilation duct is located in a corner of the room, and consequently Q = 8 (Table 15).

The constant of the room is:

$$R = \frac{100 \cdot 0{,}2}{1 - 0{,}2} = 25.$$

In view of the lack of data for this value of R on the chart (Figure 87), we perform a numerical calculation of the value of the parentheses under the sign of the logarithm in formula (159). For a distance of 1 m from the mouth of the duct, we obtain:

$$\frac{Q}{4\pi r_M^2} + \frac{4}{R} = \frac{8}{4\pi \cdot 1} + \frac{4}{25} = \frac{1}{1{,}26}.$$

The noise level is:

$$\beta_2 = 100 - 10 + 10 \log (400) - 10 \log 1.26 - 40 = 75 \text{ db.}$$

At a distance of 2 m from the mouth of the duct, the noise level, found by the same method, is equal to:

$$\beta_2 = 71 \text{ db.}$$

Closely related to the task of calculating the noise transmitted along an air duct is the task of determining the necessary value for the sound isolation of a muffler in an air duct passing from a noisy room, through a wall having a finite sound isolation value, into a quiet room. Let ZI_K and S_K represent the sound isolation and the

cross-sectional area of the air duct, and ZI_W and S_W represent the same values for the wall separating the noisy room from the quiet room (Figure 131-b).

The following sound energy penetrates through the wall into the quiet room:

$$J_W = S_W \cdot 10^{-0.1ZI_W},$$

and the following penetrates through the air duct:

$$J_K = S_K \cdot 10^{-0.1ZI_K}.$$

It is required that the energy transmitted through the air duct not exceed a level equal to $\frac{1}{k}$ of the energy penetrating through the wall,

$$S_K \cdot 10^{-0.1ZI_K} = \frac{S_W}{k} \cdot 10^{-0.1ZI_W}.$$

Whence, following transformation, we obtain the required sound isolation of the muffler in the air duct:

$$ZI_K = ZI_W - 10 \log \frac{S_W}{S_K \cdot k} \text{ db.} \qquad (160)$$

At $k = 2$, i.e. when the energy penetrating through the air duct does not exceed one-half of the energy penetrating through the wall,

$$ZI_K = ZI_W - 10 \log \frac{S_W}{S_K} + 3 \text{ db.} \qquad (161)$$

Example 44. The sound isolation of a wall of 10 m^2 area, separating a noisy room from a quiet room, is equal to 30 db. In the wall is a ventilation opening with 0.1 m^2 area. The problem is to determine the required value of sound isolation of a muffler in the air duct such that the sound isolating characteristic of the wall is not worsened.

Solution. From formula (161), it follows that:

$$ZI_K = 30 - 10 \log \frac{10}{0.1} + 3 = 13 \text{ db.}$$

In American practice (136) another formula is applied for determining the necessary value of sound isolation of an air duct connecting

a noisy room with a quiet one. Using our symbols, that formula has the following form:

$$ZI_K = \frac{1}{2}\left(ZI_W - 10 \log \frac{S_W}{S_K}\right) + 3 \text{ db.} \tag{162}$$

This formula gives lower values for the sound isolation of an air duct than does formula (161).

CHAPTER 11

MUFFLERS FOR THE AIR INTAKE AND EXHAUST OF RECIPROCATING ENGINES

41. Types of Mufflers. The Analysis of Reactive Mufflers by the Quadripole Method

Engine exhaust and air intake mufflers, similarly to ventilation system mufflers, may be divided into reactive and active types. Reactive mufflers, or acoustic filters, do not contain a special sound absorber and consist of a combination of chambers and tubes, i.e. elements of acoustic elasticity and mass. Their effect is based mainly on reflection of sound, although a certain amount of sound is absorbed on the internal metal surfaces of the elements of the mufflers and of the exhaust system of the engine.

Active exhaust mufflers require temperature resistant mufflers. Occasionally water (in so-called "wet" mufflers) or a multiplicity of metal elements (hot tubes in waste-heat boiler-type mufflers) are used as a sound absorber. No difficulty is encountered in the application of ordinary porous or fibrous organic absorbers in the air intake mufflers of engines.

Reactive mufflers were applied in exhaust silencing systems as early as the decade of 1930. The simplest method for the analysis of multiple cell mufflers of this type is that proposed in the USSR by N. N. Andreyev. This method is based on the simplifying assumption that the load resistance of each cell of the muffler (i.e. combination of one chamber and one tube) is a non-reflecting resistance. A. I. Belov showed that the error in the noise reduction of one cell under this assumption, as compared to actual conditions, does not exceed a few decibels. B. K. Shapiro (109) presented a method of analysis for the case in which the dimensions of the cells are not small in comparison to the wave length of the sound.

The design of mufflers by this method is based on the quadripole theory, which offers considerable convenience in computation and yields results which are intermediate to the formal processes of matrix

algebra. The name "quadripole" is taken from the theory of electric filters, in which a similarly rationalized process of analysis was developed.

This method can be found presented in more detail in references (43) and (109). We include here merely the final expression for the value of the noise reduction in a single cell of a muffler:

$$\Delta\beta = 20 \log \left| y + \sqrt{y^2 - 1} \right| \text{ db}, \tag{163}$$

where y is the universal parameter of the cell of the muffler, expressed in the following form through its geometric dimensions (Figure 132):

$$y = \frac{1}{2}\left[1 + \frac{1}{2}\left(\frac{S_1}{S_2} + \frac{S_2}{S_1}\right)\right]\cos k(l_1 + l_2) + \\ + \frac{1}{2}\left[1 - \frac{1}{2}\left(\frac{S_1}{S_2} + \frac{S_2}{S_1}\right)\right]\cos k(l_1 - l_2), \tag{164}$$

where S_1 and S_2 are the cross-sectional areas of the expansion chamber and connecting tube, of the noise reducing cell;

l_1 and l_2 are the lengths of the corresponding elements;

k is the wave number.

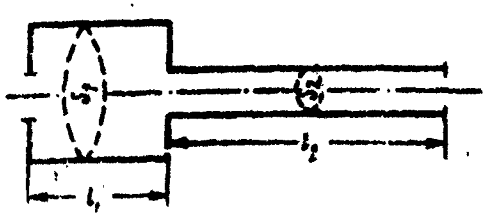


Figure 132. Cell of a reactive muffler (analysis of the muffler by the quadripole method).

The parameter y is called universal because it completely determines both the value of noise reduction and the band of frequency in which the reduction occurs. Actually, formulas (164) and (163) indicate that the noise reduction of the cell increases in proportion to an increase in the ratio of the cross-section of the chamber to the cross-section of the tube. This conclusion already has been presented earlier in connection with systems in which a sharp change in the cross-section of the sound duct is observed (formulas (87) and (154).

Furthermore, it is apparent from the above formulas that the noise reduction also is increased with an increase in the length of the chamber l_1 or in the length of the tube l_2. More detailed analysis of

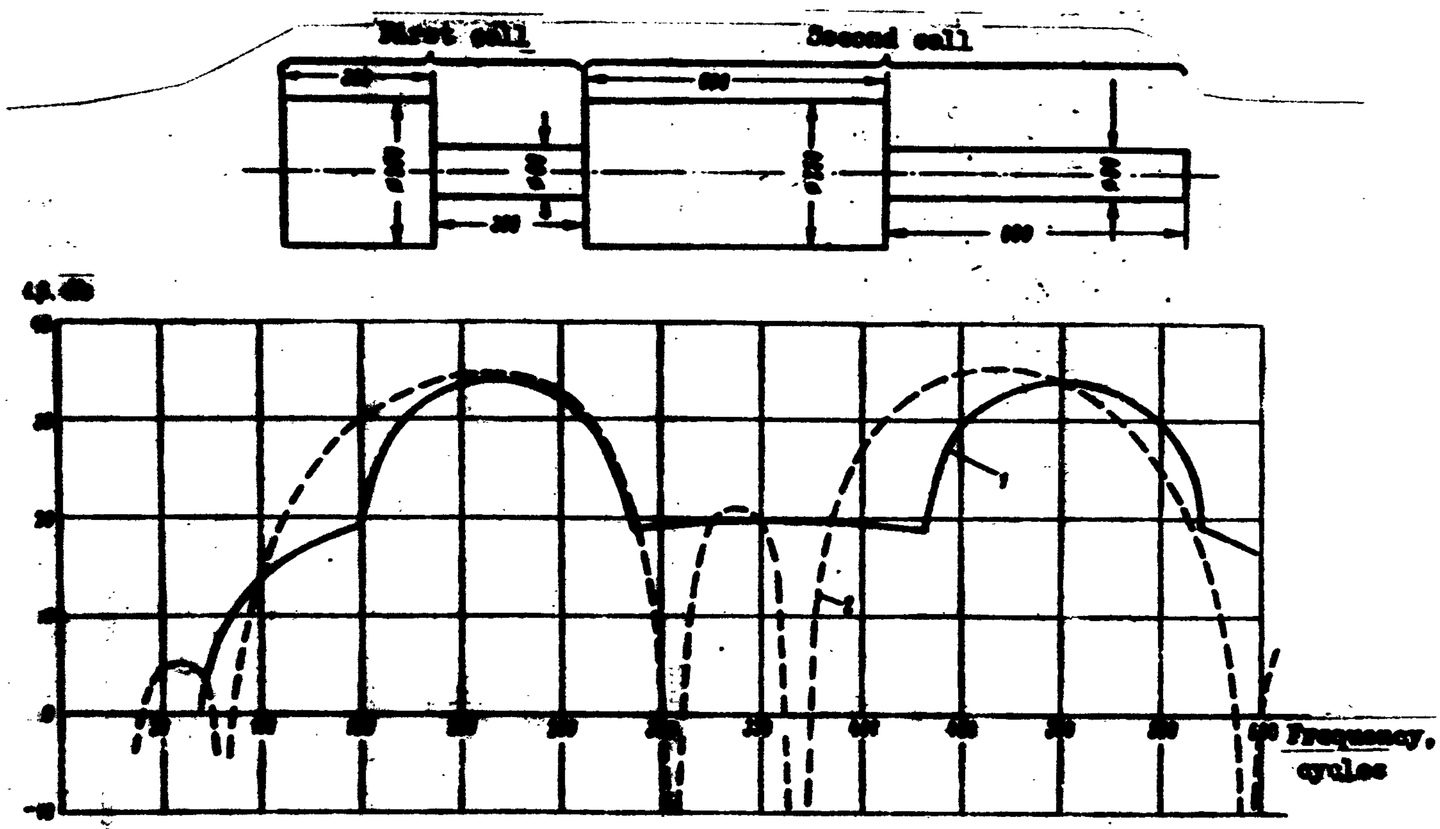


Figure 133. Frequency characteristics of the noise reduction of a two-cell muffler according to the quadripole method (curve 1) and the linked fractions method (curve 2).

formulas (163) and (164) would show that the ratio $\frac{l_2}{l_1}$ determines the width of the band of frequencies of absorption, which attains a maximum at $\frac{l_2}{l_1} = 1.$

The total reduction given by several cells of a muffler at any given frequency is determined in the quadripole simplified method by a summation of the reduction due to each of the cells at the given frequency.

Example 45. The problem is to compute the frequency characteristic of the noise reduction of the two-cell reactive muffler shown at the top of Figure 133, using the quadripole method.

Solution. To determine the frequency characteristic of the coefficient of noise reduction of the first cell of the muffler, we find the value of the universal parameter Y, as given in formula (163).

We take the dimensions from Figure 133. The temperature of the gases in the muffler is approximately 300° C. The speed of sound at this temperature may be taken as equal to $c = 4.1 \cdot 10^4$ cm/sec (109). Thus we have:

$$Y = \frac{1}{2}\left[1 + \frac{1}{2}\left(\frac{25^2}{8^2} + \frac{8^2}{25^2}\right)\right] \cos \frac{2\pi}{4{,}1\cdot 10^4} f(30+30) +$$

$$+ \frac{1}{2}\left[1 - \frac{1}{2}\left(\frac{25^2}{8^2} + \frac{8^2}{25^2}\right)\right] \cos \frac{2\pi}{4{,}1\cdot 10^4} f(30-30) \approx 2{,}96 \cos (0{,}5f)^\circ - 1{,}96.$$

We determine the value of the reduction for a frequency, such as 300 cycles:

$$\Delta\beta_1 = 20 \log \left[(2{,}96 \cos (0{,}5\cdot 300)^\circ - 1{,}96) + \right.$$

$$\left. + \sqrt{[2{,}96 \cos (0{,}5\cdot 300)^\circ - 1{,}96]^2 - 1}\right] \approx 19 \text{ db}.$$

For the second cell, we have, similarly:

$$Y_2 = \frac{1}{2}\left[1 + \frac{1}{2}\left(\frac{25^2}{8^2} + \frac{8^2}{25^2}\right)\right] \cos \frac{2\pi}{4{,}1\cdot 10^4} f(60+57) +$$

$$+ \frac{1}{2}\left[1 - \frac{1}{2}\left(\frac{25^2}{8^2} + \frac{8^2}{25^2}\right)\right] \cos \frac{2\pi}{4{,}1\cdot 10^4} f(60-57) \approx 2{,}96 \cos f^\circ - 1{,}96.$$

The reduction due to this cell at a frequency of 300 cycles is equal to zero because $\mathcal{Y}_{2f\,=\,300\ \mathrm{cycles}} < 1$. The total reduction at a frequency of 300 cycles is:

$$\Delta\beta_{\Sigma} = \Delta\beta_1 + \Delta\beta_2 = 19 + 0 = 19\ \mathrm{db}.$$

The value of the noise reduction for the remaining frequencies is found in the same way. The frequency characteristic of the noise reduction which is obtained is shown in Figure 133 (curve 1).

42. The Analysis of Reactive Mufflers by the "Method of Linked Fractions" /metod tsepnykh drobey/

V. S. Nesterov (80) proposed use of the "method of linked fractions" for the analysis of multi-cell acoustic circuits of the type found in multiple-layered resonance sound mufflers. The linked fractions method was applied independently of the latter by O. V. Petrova and V. P. Terskikh in the analysis of marine engine exhaust mufflers. According to this method, in which, contrary to the method described in the preceding paragraph, no assumptions are made concerning the mutual compatibility of the elements, the individual elements of the exhaust gas system (tubes and chambers of the muffler and manifolds) are replaced by combinations of various quantities, namely by the equivalent stiffness $H_i^э$ and the equivalent compliance $E_i^э$, which represent the inertial and elastic properties of the elements (Figure 134, a and b). For the purpose of rendering the system symmetrical, fictitious elements with zero compliance are introduced at the functions of elements (Figure 134-c).

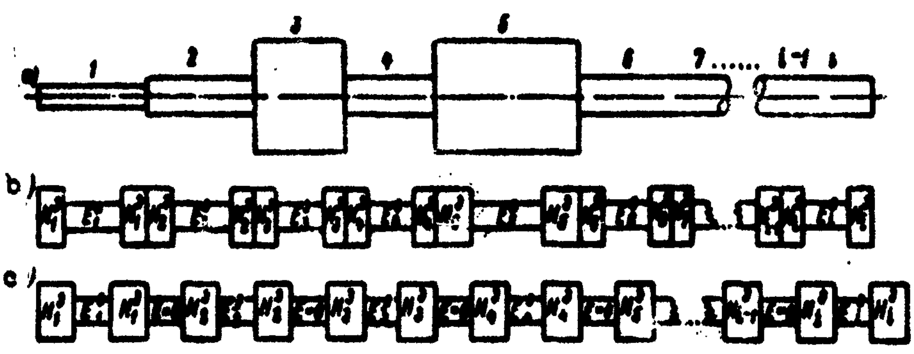


Figure 134. Reduction of a reactive muffler to a system of stiffness and compliance quantities (in the analysis of a muffler by the linked fractions method).

The design of mufflers according to the linked fractions method is performed as follows. First of all the values of the mass μ and the compliance ε of the individual elements of the system are determined:

$$\mu = \frac{l\rho}{S}; \quad \varepsilon = \frac{Sl}{\rho c^2}, \tag{165}$$

where l and S are the length and cross-sectional area of the muffler element;

ρ and c are the air density and speed of sound within the muffler at the given temperature.

For convenience, the computation is performed in relative units, i.e. we determine

$$m = \frac{\mu}{\mu_0}; \quad e = \frac{\varepsilon}{\varepsilon_0}, \tag{166}$$

where μ_0 and ε_0 are arbitrary values of acoustic mass and compliance.

The values of the equivalent stiffnesses and the equivalent compliances of the elements of the system are found from the following expressions:

$$H_i^* = \frac{1}{l_i}\left(-\sqrt{m_i l_i \Delta}\cdot\tan\frac{\sqrt{m_i l_i \Delta}}{2}\right); \tag{167}$$

$$E_i^* = l_i\frac{\sin\sqrt{m_i l_i \Delta}}{\sqrt{m_i l_i \Delta}}; \tag{168}$$

here Δ is a particular frequency parameter, equal to:

$$\Delta = 4\pi^2 f^2 \mu_0 \varepsilon_0. \tag{169}$$

It may be shown that the ratio of the amplitudes of vibration of the gas at the inlet and outlet of the gas exhaust system, $\frac{\xi_{in}}{\xi_{out}}$, is equal to:

$$a_{in} = \frac{\xi_{in}}{\xi_{out}} = H_i^{\circ}\left(E_i^{\circ} + \frac{1}{H_i^{\circ}}\right)\left(H_{i-1}^{\circ} + H_i^{\circ} + \frac{1}{E_i^{\circ} + \frac{1}{H_i^{\circ}}}\right) \times \cdots$$

$$\cdots \times \left(E_1^{\circ} + \cfrac{1}{H_1^{\circ} + H_2^{\circ} + \cfrac{1}{E_2^{\circ} + \cfrac{1}{H_2^{\circ} + H_3^{\circ} + \vdots + \cfrac{1}{E_i^{\circ} + \cfrac{1}{H_i^{\circ}}}}}}\right). \tag{170}$$

The noise reduction of the muffler at the given frequency is:

$$\Delta\beta = 20 \log \left(\frac{\xi_{in}}{\xi_{out}}\right) \text{ db.} \tag{171}$$

The design of a muffler according to the linked fractions method is reduced to the performing of several typical operations, the essence of which is explained by the table of Figure 135, which has been constructed for designing a four-element muffler. Completion of the table in this outline begins with line 1, in which the values of stiffness and compliance found according to formulas (167) and (168) for each of the elements of the system are entered. The further sequence of computation is indicated by arrows in the outline. The order of progression of the computations is from right to left in the diagram.

Muffler element number→	1			2				3				4		
1	E_1°	H_1°	0	H_2°	E_2°	H_2°	0	H_3°	E_3°	H_3°	0	H_4°	E_4°	H_4°
2										$H_4^{\circ}+\frac{1}{E_4^{\circ}+\frac{1}{H_4^{\circ}}}$		$\frac{1}{E_4^{\circ}+\frac{1}{H_4^{\circ}}}$	$\frac{1}{H_4^{\circ}}$	
3	(8)	(7)			(6)	(5)			(4)	$H_3^{\circ}+H_4^{\circ}+\frac{1}{E_4^{\circ}+\frac{1}{H_4^{\circ}}}$ (3)		$H_4^{\circ}+\frac{1}{E_4^{\circ}+\frac{1}{H_4^{\circ}}}$	$E_4^{\circ}+\frac{1}{H_4^{\circ}}$ (2)	H_4° (1)

Figure 135. Table for the design of a muffler by the linked fractions method.

Methods for computation of linked fractions are described in the monograph of V. P. Terskikh (103). The quantity $\frac{\xi_{in}}{\xi_{out}}$ is determined by cross-multiplication of the values (1), (2), ..., (8) in parentheses in line 3 of the table.

Despite a certain cumbersomeness, the design of reactive mufflers according to the linked fractions method is valuable in that it enables a precise determination of the location of faults in the frequency characteristic of the sound isolation of mufflers, and may be applied to exhaust gas systems as a whole. Analysis of the nature of the change in the magnitude of the relative amplitude while traversing the system (the products of the figures in parentheses in Figure 135) enables the most rational dimensions of each element of the system to be determined.

Example 46. The problem is to compute the frequency characteristic of the noise reduction for the reactive absorber indicated in Example 45, by the linked fractions method.

Solution. To compute a_{1n} we first find the value of the mass and compliance of each of the elements of the muffler. The values of the speed of sound in the muffler and the air density, as in the preceding example, are taken with the assumption that the air temperature in the muffler is approximately 300° C. For the first element of the muffler (chamber diameter 250 mm) we obtain:

$$\mu_1 = \frac{l_1\rho}{S_1} = \frac{30\cdot 6,15\cdot 10^{-4}}{0,785\cdot 25^2} = 3,77\cdot 10^{-5};$$

$$\varepsilon_1 = \frac{S_1 l_1}{\rho c^2} = \frac{0,785\cdot 25^2\cdot 30}{6,15\cdot 10^{-4}\cdot 4,1^2\cdot 10^8} = 1,46\cdot 10^{-2}.$$

Converting to relative units, using the arbitrary form $\mu_0 = 40.2\cdot 10^{-5}$, $\varepsilon_0 = 0.132\cdot 10^{-2}$, we obtain:

$$m_1 = \frac{\mu_1}{\mu_0} = \frac{3,77\cdot 10^{-5}}{40,2\cdot 10^{-5}} = 0,094; \quad e_1 = \frac{\varepsilon_1}{\varepsilon_0} = \frac{1,46\cdot 10^{-2}}{0,132\cdot 10^{-2}} = 11,1.$$

The results of similar computations for other elements of the muffler are shown in the following table.

Elements	Values			
	μ	ε	m	e
2	$36,9\cdot 10^{-5}$	$0,15\cdot 10^{-2}$	0,92	1,14
3	$7,54\cdot 10^{-5}$	$2,92\cdot 10^{-2}$	0,19	22,1
4	$74\cdot 10^{-5}$	$0,3\cdot 10^{-2}$	1,84	2,29

Let us compute the value of the particular frequency parameter for a frequency of 300 cycles.

$$\Delta = 4\pi^2 (300)^2 \cdot \mu_0 \cdot \varepsilon_0 = 1,865$$

We find the values of the equivalent stiffness and compliance of the first element of the muffler according to formulas (167) and (168):

$$H_1^* = \frac{1}{11,1}\left(-\sqrt{0,094 \cdot 11,1 \cdot 1,865} \cdot \tan \frac{\sqrt{0,094 \cdot 11,1 \cdot 1,865}}{2}\right) = -0,105;$$

$$E_1^* = 11,1 \frac{\sin \sqrt{0,094 \cdot 11,1 \cdot 1,865}}{\sqrt{0,094 \cdot 11,1 \cdot 1,865}} = 7,85.$$

The values of the same quantities for other elements of the muffler are:

Elements	Values	
	H^*	E^*
2	−1,02	0,808
3	−0,695	2,79
4	−7,45	0,261

We insert the obtained values into the outline of Figure 135 and, proceeding from right to left, determine sequentially, by the linked fraction method, the values of the intermediate quantities in the table as follows:

1			2				3		
7,85	−0,105	0	−1,02	0,808	−1,02	0	−0,695	2,79	−0,695
0,697	1,14		2,16	−0,344	−1,885		−1,19	−3,63	0,42
(8,817)	(1,035)		1,14	(0,464)	(−2,905)		−1,885	(−0,84)	(−0,275)

(Table continued)

4			
0	—7,45	0,261	—7,45
	7,87	—0,134	←
	0,42	(0,127)	(—7,45)

According to the outline of Figure 135 the value α_{in} is obtained through multiplication of the values in the parentheses:

$$\alpha_{in} = (-7{,}45)(0{,}127)(-0{,}275)(-0{,}84)\times$$
$$\times(-2{,}905)(0{,}464)(1{,}035)(8{,}817) = 2{,}69.$$

The noise reduction of the muffler at a frequency of 300 cycles is equal to:

$$\Delta\beta = 20 \log \cdot \alpha_{in} = 8.6 \text{ db.}$$

The noise reduction is computed similarly for other frequencies. The frequency characteristic of a muffler obtained in this way is shown in Figure 133 (curve 2). Comparing it with curve 1, it is seen that the limits of the frequency zones of high damping determined by the two methods practically coincide. The same may be said to be true of the zone of low damping in the range of medium frequency. However, as mentioned in the foregoing, computation of a muffler by the simplified method of quadripoles does not reveal the frequency faults of sound isolation in the frequency characteristic of the noise reduction of the muffler. Because of this, in the case of intense components at individual frequencies in the spectrum of noise of the exhaust, it is advantageous, in addition to analysis of the muffler by the quadripole method, to supplement this analysis with a calculation according to the linked fractions method. The use of computing machines greatly facilitates the latter analysis.

In view of the complexity of a muffler as an acoustic system it must be considered only as a first stage of actual designing. The final data for a muffler are determined after testing a model of the muffler and correction (if necessary) of its design parameters. Laboratory testing of the acoustic characteristics of exhaust mufflers is performed with the aid of artificial sources of sound and is similar to the testing of ventilation duct mufflers. Practical evaluation of

the acoustic properties of a muffler is performed through tests on an actual engine.

43. The Design and Acoustic Properties of Reactive and Active Mufflers

Figure 136 shows several designs of mufflers for low power marine engines. The average noise reduction of the mufflers in the sonic frequency range also is given, as the power of the engine is changed from 25 to 100 percent of nominal capacity.

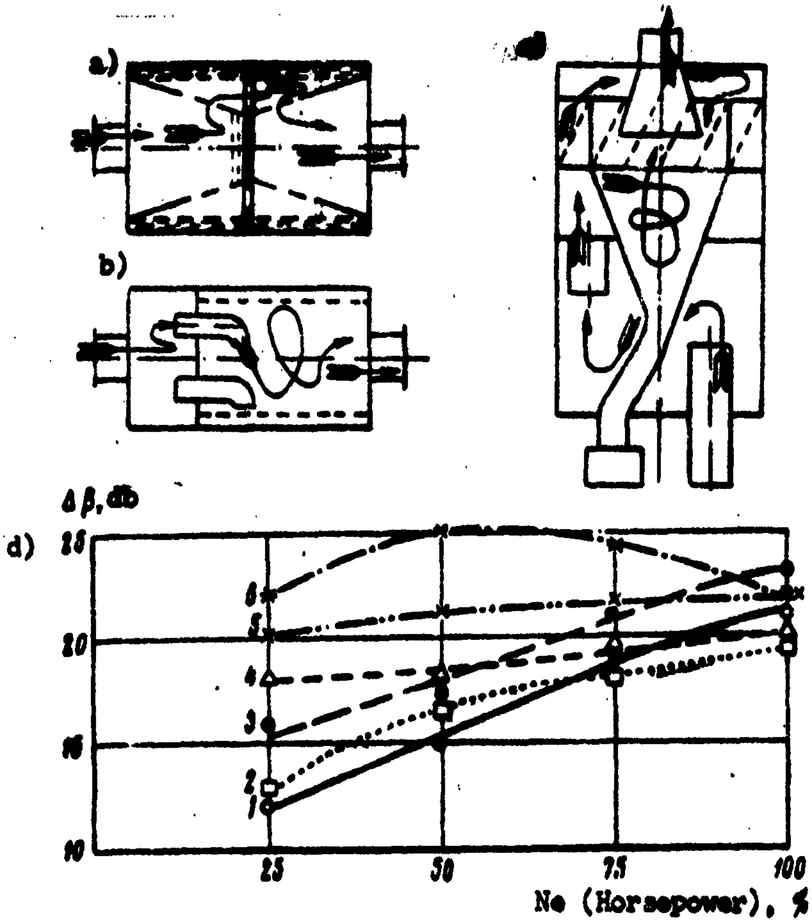


Figure 136. Designs of mufflers, and their noise reducing effect as a function of the type of engine and its power.
1 - Muffler a, engine type 2 Ch 8.5/11; 2 - Muffler b, engine type 2 Ch 13/18; 3 - Muffler 1, engine type 4 Ch 8.5/11; 4 - Muffler b, engine type 4 Ch 13/18; 5 - Same, engine type 2 Ch 13/18; 6 - Same, engine type 3D6.

The designs shown in Figure 136, b and c are reactive absorbers in which, however, the exhaust gases are given a rotational motion, primarily to remove solid particles (this also may be achieved through the introduction of ultrasonic vibrators in the exhaust system, or in the muffler, itself). The muffler design shown in Figure 136-a includes a sound absorber in addition to reactive elements. The former may consist of very fine steel wire gauze, which is obtained by a metallurgical operation involving a special process.

As may be seen, the noise reduction varies from 13 or 15 db, to 23 or 25 db, and depends not only upon the design of the muffler, but upon the type of engine, as well. This is due to differences in the design and dimensions of the exhaust manifold, and again emphasizes the approximate nature of computation according to the quadripole method, in which the degree of lack of compatibility of the cells of the muffler with each other and with the elements of the exhaust system of the engine is not taken into account. With the exception of a single case, the acoustic effect of the muffler increases with the increase in the power developed by the engine. This is explained mainly by the fact that the volumetric speeds in the gas stream increase with the increase in engine power, and consequently the friction loss in the muffler also increases.

Figure 137-a shows the design of a "wet" muffler, in which noise reduction is due to the absorption of sound by drops of water formed by the passage of a mixture of water and exhaust gases through cylindrical tubes with perforated walls. This muffler also is a good spark arrestor. A diagram of the installation of the muffler on the river passenger launch "Moskvich" is shown in Figure 137-b. In conjunction with an underwater gas exhaust system, the "wet" muffler has almost completely eliminated the exhaust noise on the deck of the launch (48).

An excellent practical solution to the problem of reducing exhaust noises is the use of waste-heat boilers, which are present in some ships and which function on the exhaust gases of the engine, as a muffler. The cluster of tubes in the boiler reduces noise at high frequencies, and the expansion chambers, fitted before and after the boiler, are adjusted for reduction of low frequency noise. The total reduction of noise (over the spectrum) reaches 18 db (74).

Figure 138 shows cross-sections of an air intake muffler tested by the present author for medium-power diesels. The basis of the muffler is a tube with perforated walls, under which is a sound absorber (glass wool, mineral wool, cotton wadding, plastic foam). The reduction of sound in a tube of this type is calculated in the same manner as that in ventilation system mufflers. Additional reduction of sound, attaining 4 or 5 db, occurs as a result of turning the air stream and the absorption of sound by the dust-protection mesh and sound absorber

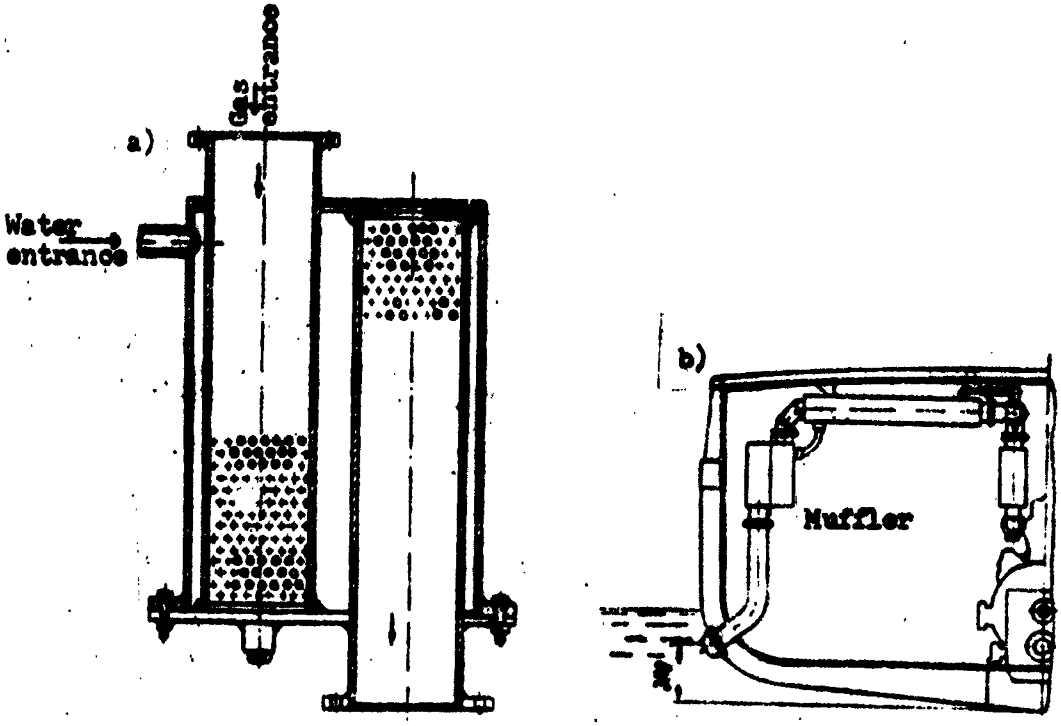


Figure 137. A "wet" muffler (a) and its installation on a river passenger launch (b).

on the internal surface of the cover (the gap between the cover and the body of the absorber may be regulated, to enable selection of the desired degree of silencing at any given back-pressure).

A reactive filter, consisting of a space of toroidal shape and a perforated cover is connected in parallel to the active muffler (the bottom wall of this space may be solid or may be perforated). The filter is adjusted to the frequency of the "siren" sound of the air blower (cf. Chapter 17). The total reduction of noise of the air blower after installing the muffler can reach 20 to 25 db.

During World War II, nozzle-type air intake mufflers were applied on the high-power diesels of German ships, the characteristic peculiarity of which was a series of expansion nozzles at the intake end of the muffler (141). If the speed of flow of the gas in the throat of the nozzle is made equal to the speed of sound, then the sound, in principle, cannot leave the muffler. In practice, however, for the purpose of preventing losses due to friction of the gas stream against the wall of the nozzle, a considerably lower speed of flow is

necessary. Nevertheless, the muffler provides quite adequate reduction of vibration over a broad range of frequencies.

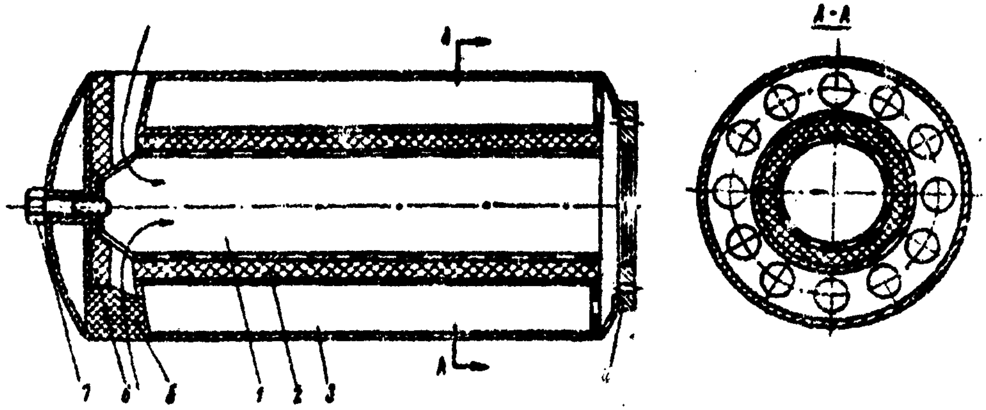


Figure 138. Air intake muffler with sound absorber.
1 - Tube with perforated walls; 2 - Sound absorber; 3 - Parallel-arranged chamber; 4 - Flange for attachment to engine; 5 - Protective grating; 6 - Absorber on cover; 7 - Screw for regulating air intake aperture. Arrows indicate direction of motion of air.

44. Hydraulic Resistance of Mufflers

A muffler must be designed in a manner to ensure that it exerts minimal resistance to the constant flow of gas within it, i.e. that the drop in pressure should be as low as possible. For a value of pressure drop Δp of the gas stream in a round tube, an expression taken from reference (43), with a certain amount of modification, holds true:

$$\Delta p = \mathrm{const}\,\frac{l\rho v^2}{\sqrt[4]{\mathrm{Re}\cdot d^5}}, \qquad (172)$$

where l and d are the length and diameter of the tube;

ρ and v are the density of gas and volumetric velocity of its motion;

Re is the Reynolds number.

The great dependence of the transmission of pressure drop upon the diameter of the tube merits attention. Because of this, increasing the acoustic effect of the muffler by means of a reduction of the tube

diameter of the muffler is disadvantageous, because it results in a considerable loss of power in the engine. It is more rational to increase the length and cross-sectional area of the chamber, and also the length of the tubes. These factors show the difficulty of creating small-size reactive mufflers with a high noise reduction and low back-pressure. Thus the trend toward utilization of all the free volume of the engine room (along the bulkheads, at the overhead, etc.) for placement of mufflers is completely justified from this point of view.

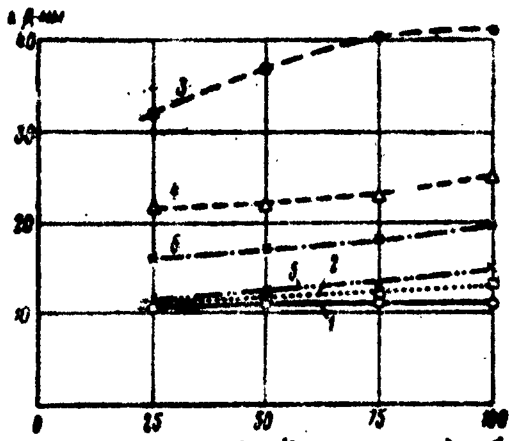


Figure 139. Back pressure of mufflers as a function of type of engine and the power developed by it. Muffler design and notations of curves as in Figure 136.

Figure 139 shows the back pressure of the muffler designs described earlier, as a function of the engine power. The magnitude of the back pressure varies between 10 and 40 mm Hg, and, just as for the magnitude of the noise reduction, depends upon the type of engine and the power developed by it. The variation in back pressure is mainly a function of the volumetric velocity in the gas exhaust system of the engine.

There are indications (43) of the possibility of reducing the back pressure of mufflers through the inclusion of parallel chambers of large volume and the selection of the length of the exhaust pipe in such a manner as to ensure a condition of resonance in the operation of the muffler in conjunction with the engine.

CHAPTER 12

THE ISOLATION OF SONIC VIBRATION

45. The Vibration Isolating Effect of Elastic Materials

Ship hull structures conduct sound better than the structures of ordinary buildings. In Table 7 it was seen that, at least in small ships, airborne noise in rooms is related in its origin to sonic vibrations emanating from the region of the engine room.

Because of this, the isolation of sonic vibration is of primary importance to the control of noise in ships. It is accomplished by mounting the sources of vibration on sound isolating mounts. It is evident that this must be accompanied by isolation and absorption of airborne sound.

The results of experiments conducted by King (141), who investigated the reduction of gear noise by means of the simultaneous and separate application of sound-, and vibration-isolating treatments, are revealing. The simultaneous application of a sound-protecting housing and sound isolating mounts enabled the loudness of the sound of the gears to be reduced by 28 phons. When each of these means was applied separately, the acoustic effect did not exceed 3 or 4 phons. This remarkable effect of sound isolation mounts in relation to noise in the compartment of the source is characteristic only of the case in which the direct sound from the machine is measured together with the sound radiated into the room by the vibrating walls. Usually, direct sound predominates in the room of the source (if there is no resonance of parts of the boundary structures), and the installation of vibration isolation mounts has almost no effect on the noise level in that compartment. However, the effect of sound isolating mounts or elastic gaskets under the machine is fully manifest in adjacent compartments, in which noise is determined by the sonic vibrations of the walls.

For the purpose of clarification of the basic laws of isolation of sound waves by means of elastic materials we shall consider the idealized case of an infinite metallic sound conductor, the material of which has wave resistance $(\rho c)_1$. The equivalent wave resistance of the material of an insert or gasket /prokladka/ is ρc (Figure 140-a).

Because materials with a wave resistance larger than $(\rho c)_1$ do not exist, the only possibility of attaining isolation of vibration with inserts or gaskets consists of the use of materials in which $\rho c \ll (\rho c)_1$.

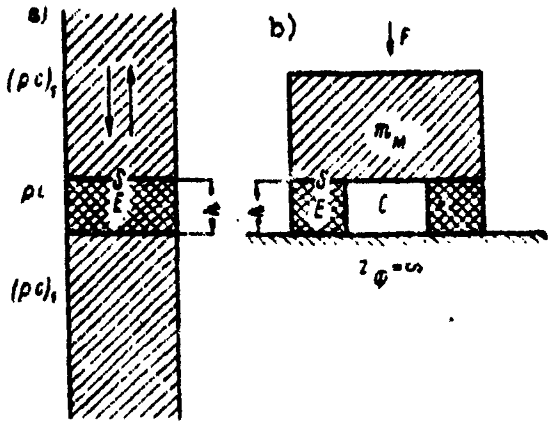


Figure 140. Elastic sound isolating gaskets placed in an infinite sound conductor (a) and under a machine (b).

Utilizing a method similar to that employed in the derivation of the expression for the isolation of airborne sound by partitions having appreciable mass (Paragraph 24), we may obtain an expression for the isolation of a longitudinal wave in a metallic sound conductor due to the insertion of an elastic gasket:

$$VI = 10 \log \left[1 + \left(\frac{\pi f (\rho c)_1}{\frac{E}{h}} \right)^2 \right] \text{db.} \tag{173}$$

Here $(\rho c)_1$ is the wave resistance of the metallic sound conductor;

h and E are the thickness of the gasket and some equivalent modulus of elasticity of its material;

f is the frequency of vibration.

Taking account of the fact that $\frac{h}{E} = K$ corresponds to the compliance of the gasket per cm^2 according to Hook's law, and that for sufficiently large values of frequency the numerical terms may be ignored in comparison with the second term, we present the above formula in the form:

$$VI = 20 \log (fK) - \text{const db.} \tag{174}$$

Comparing this expression with formula (103), it is apparent that in the isolation of "material" sound, the compliance of a gasket fulfills the same role as the mass of the wall in the isolation of airborne sound.

Flexural vibrations of low and medium sonic frequency are isolated less by an elastic gasket than are longitudinal vibrations (124).

Friction in the material of the insert or gasket plays a definite role in the isolation of "material" sound. The positive value of friction is evident at frequencies in which a whole number of sonic half-waves are encompassed within the thickness of the insert. This wave resonance is analogous to the resonance of the air space in double partitions isolating airborne sound. The resonance frequencies may be found from the condition:

$$h = n\frac{\lambda}{2} = n\frac{c}{2f_r};\ n = 1, 2, 3 \ldots \tag{175}$$

where c is the velocity of propagation of waves in the insert.

The frequency of the first resonance is equal to:

$$f_{r1} = \frac{c}{2h}. \tag{176}$$

The vibration-conducting properties of inserts at wave resonance frequencies are determined entirely by the magnitude of the mechanical losses in them. The greater these losses, the less is the deterioration of the vibration isolation at resonance. At the same time, however, the presence of losses causes a slight increase in vibration transmission outside the regions of resonance (cf. Figure 141).

Usually in elastic materials the forces of internal friction at high frequencies are fairly high, and to a greater or lesser extent only the first wave resonance may appear. This must be taken into account when "material" sound has very intense components, such as in the magnetic hum of electric motors.

The frequency of the first resonance of an elastic insert is lower, the thicker the insert. On this basis several German scientists do not recommend the use of thick vibration isolation mounts of elastic material. Experience, however, substantiates the considerable useful effect of the forces of internal friction in such materials, and indicates that the sound isolation of thick inserts is considerably

greater in the sonic frequency range than that of thin inserts, which also follows from formula (173).

The model of an infinite sound conductor (Figure 140-a), which explains several factors in the phenomenon of vibration isolation, holds, for example, in the case of a long metal shaft, into which sound isolating couplings or gaskets have been introduced. A vibration-isolated machine at low and medium sonic frequencies may be depicted approximately by the diagram of Figure 140-b. Here F is a force of varying frequency arising in a machine in the course of its operation, m_M is the mass of the machine, and C is the total stiffness of the elastic mounts, which in the first approximation is equal to:

$$C = \frac{ES}{h}, \tag{177}$$

where h, S and E are the thickness, total supporting surface and equivalent dynamic modulus of elasticity of the material of the mounts, respectively. For the sake of simplicity let us assume that friction is absent in the material, and that the mass of the foundation under the vibration-isolated machine is infinitely great (impedance of the foundation $z_f = \infty$).

The impedance of a system consisting of mass m_M and elasticity C is equal to (formula 28):

$$z = j\left(\omega m_M - \frac{C}{\omega}\right).$$

The value of the vibrational velocity of a mass under the effect of a force F (same as formula 26) is:

$$\dot{y} = \frac{F}{z} = \frac{F}{j\left(\omega m_M - \frac{C}{\omega}\right)}.$$

The force transmitted by elastic mounts to the foundation is:

$$\dot{F}_f = \dot{y} z_c,$$

where z_c is determined from formula (30). We obtain:

$$F_{\Phi} = -\frac{F j \frac{C}{\omega}}{j\left(\omega m - \frac{C}{\omega}\right)} = \frac{F}{1 - \frac{\omega^2 m_M}{C}}. \qquad (178)$$

The ratio $\frac{C}{m_M}$ is identical to the square of the circular frequency of free vibrations of a machine on vibration isolation mounts (formula 9):

$$\omega_0^2 = \frac{C}{m_M}.$$

When the mass is rigidly attached to the foundation, the entire vibratory force F is transmitted to the foundation. The ratio $\frac{F}{F_{\Phi}}$, thus determines the vibration isolation effect of the elastic mounts. From formula (178), it is equal to:

$$\frac{F}{F_{\Phi}} = 1 - \left(\frac{\omega}{\omega_0}\right)^2. \qquad (179)$$

The reciprocal of $\frac{F}{F_{\Phi}}$ is known in the theory of vibrations as the coefficient of transmission.

The vibration isolation is:

$$VI = 20 \log\left|1 - \left(\frac{\omega}{\omega_0}\right)^2\right| = 20 \log\left|1 - \left(\frac{f}{f_0}\right)^2\right| \text{db}. \qquad (180)$$

In this expression the ratio of the circular frequencies is replaced by the ratio of frequencies. The logarithmic expression is taken in absolute value because its sign determines only the phase of the vibratory force; the phase does not have any significance in a computation of the degree of vibration isolation.

At $\frac{f}{f_0} > 3$, we have, approximately:

$$VI_{\frac{f}{f_0} > 3} \cong 40 \log\left(\frac{f}{f_0}\right) \text{db}. \qquad (181)$$

In Figure 141 the solid curve indicates the vibration isolation of elastic mounts without friction, plotted according to the two

last-named formulas (the limits of applicability of each of the formulas are indicated on the graph). It is apparent that up to a frequency equal to $\sqrt{2} f_0$, vibration isolation is negative, i.e. the vibratory force at the foundation is increased in comparison with rigid attachment of the machine. The isolating effect of the vibration mounts is observed only at higher frequencies.

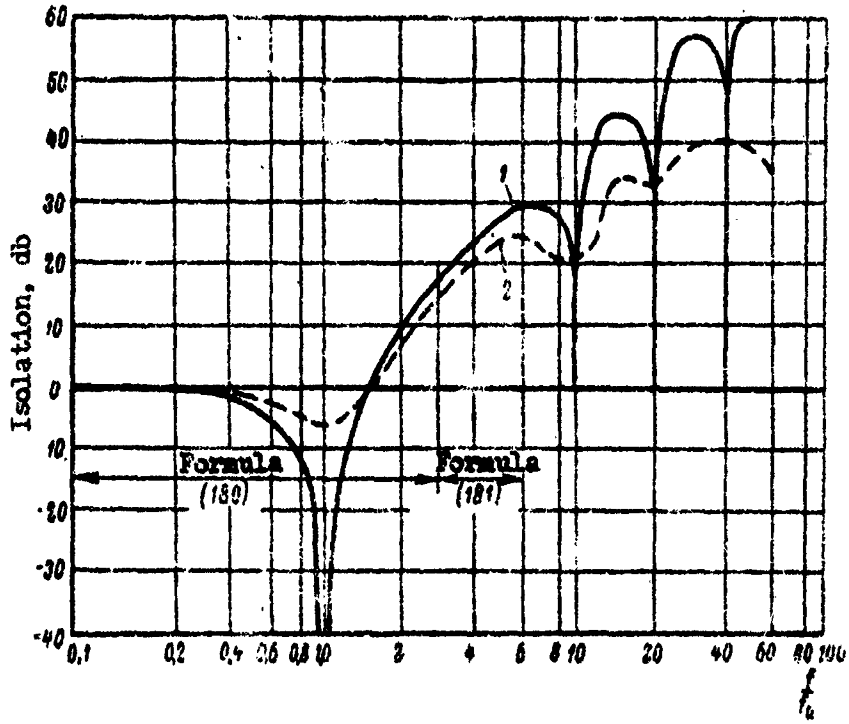


Figure 141. Frequency function of the vibration isolation of elastic mounts without friction (1) and with friction (2).

In addition to the fundamental resonance of the system (machine and elastic mounts), the wave resonance of the mounts, mentioned above, also may appear. We may repeat that as mentioned above, this resonance is almost unnoticeable in real mounts with friction (curve 2, Figure 141).

Example 47. The frequency of free vertical vibrations of a machine on elastic mounts was, due to the installation of softer elastic mounts, reduced from $f_{01} = 20$ cycles to $f_{02} = 10$ cycles. By what amount was the vibration isolation increased at a frequency of $f = 60$ cycles?

Solution. Because the ratio $\frac{f}{f_0}$ is greater than 3, we use formula (181).

The vibration isolation for the initial installation of vibration mounts is:

$$VI_1 = 40 \log \left(\frac{f}{f_{01}}\right) \text{db}.$$

The vibration isolation with softer vibration mounts is:

$$VI_2 = 40 \log \left(\frac{2f}{f_{01}}\right) \text{db}.$$

The difference in vibration isolation values is:

$$VI_2 - VI_1 = 40 \log 2 = 12 \text{ db}.$$

This example indicates the advantage, from the point of view of acoustics, of reducing the frequency of free vibrations of machines mounted on vibration isolation mounts. At the present time vibration isolation mounts with free vibration frequencies of 10 and even 5 cycles are already being used in ships. At these values of f_0, the isolation of vibration in a direction perpendicular to the plane of the foundation begins to take effect at frequencies from 7 to 14 cycles.

46. Materials for Vibration Isolation Mounts

In principle, any elasto-viscous material may be used as the isolating material of a vibration isolation mount: rubber, cork, felt, etc. Rubber is used most often under shipboard conditions, not only because of its very good vibration isolation properties, but also because of other advantages, such as its constancy of elastic characteristics with the passage of time, the ease of giving any desired shape to the material, resistance to humidity, molecular stability, and great strength of adherence to metal (after vulcanization). The disadvantages of rubber, such as susceptibility to the effects of oil, benzine and ozone (produced by the arcing of brushes of electric motors) may be countered by coating its surface with suitable lacquers and the selection of oil-, and benzine-resistance types of rubber. Experience indicates that 4 or 5 years of service on shipboard is common for rubber vibration mounts.

Table 21 (129) lists materials used in vibration isolating devices. The usual loadings placed on the material, and the values of the static and dynamic moduli of elasticity of the materials, E_{st} and E_{dyn}, are indicated. In elastic materials the latter always exceed

Table 21

Properties of Mounts Made of Various Vibration Isolating Materials, Subjected to Different Loadings

Mount Material	Thickness (under load) cm	Loading, kG/cm^2	E_{ct} kG/cm^2	E_{dyn} kG/cm^2	$\frac{E_{dyn}}{E_{ct}}$	f_0
Rubber of Average Stiffness	2,5	0,7 3,5	60—80	110 120	1,3 1,5	37 18
Soft Rubber	2,5	3,5	24	31	1,3	8
Soft Cork	3,2 2,2 1,6 1,4	1 2 3 4	10 12 20 30	60 80 160 280	6 6,7 8 9	25 25 30 35
Fiberglass (sheets)	3,8 3,5 3,1	0,05 0,1 0,2	0,8 0,8 1,1	1 1,6 3,2	1,2 2 2,9	11 10 11
Wood Fiber Material (mats)	1,8 1,6	0,05 0,1	0,45 0,77	0,54 1,0	1,2 1,3	12 10

the static modulus, a matter which is explained by the following simple consideration.

The static stiffness of any given elastic material is equal to:

$$C_{cr} = \frac{F}{\Delta_{cr}},$$

where F is the static force applied;

Δ_{cr} is the deformation of the material.

During the action of a periodic deforming force on an elasto-viscous material, the material cannot follow this force due to a property of such materials known as the so-called lag effect. At the moment of time corresponding to maximum force, the deformation is equal to $\Delta_{dyn} = \Delta_{cr} - a$, where a is any positive number. It is not difficult to see that the corresponding "instantaneous" dynamic stiffness (elasticity) is greater than the static value.

$$C_{dyn} = \frac{F}{\Delta_{ct} - a} .$$

The degree of excess of C_{dyn} over C_{cr}, or which is the same, of E_{dyn} over E_{cr}, depends upon the type of material. The ratio $\frac{E_{dyn}}{E_{cr}}$ must be taken into consideration in computing the natural frequencies f_0 of the machine on its mounts. At frequencies up to 50 or 60 cycles, above which the values of the natural frequencies of vibration isolating devices do not extend, this ratio varies for rubber between 1.1 and 1.7 or 2, and is several units in the case of cork.

Table 22, derived by the present author, describes the vibration isolating properties of various materials. Cubes 40 x 40 x 40 mm in size were placed under noisy machines. The specific loading on the mount material was 2 kG/cm^2.

The data of the table substantiate the fact that the best vibration isolation is provided by mounts of soft rubber. In the given experiment the average vibration isolation of this material at a frequency range of 100 to 700 cycles was 20 db Mounts made of pressed cork offer considerably less isolation. The vibration isolation of wooden blocks, on which machinery occasionally is mounted in ships, does not exceed 4 to 7 db, which corresponds to a noise reduction in an adjacent room of 1 -1/2-fold.

Table 22

Vibration Isolation of Mounts 40 x 40 x 40 mm, Made of Various Materials

Material	Average vibration isolation in the frequency range 100 - 700 cycles, db
Soft rubber (Shor hardness 30)	18 - 20
Semi-hard rubber (Shor hardness 55)	14 - 16
Pressed cork	11 - 13
Oak, birch (with grain)	4 - 6
Same, across grain	5 - 7

Considerable vibration isolation may be obtained from steel springs, in strip or cylindrical form. This is evident from an examination of Table 7. However, steel springs have a low friction loss, and for the purpose of avoiding transmission of vibration at numerous wave resonance frequencies it is advantageous to introduce friction, in the form of surface (in multi-layer springs) or internal elements, or in the form of supplementary rubber strips of some kind of vibration absorbing coating.

Let us consider the variations in the acoustic effect of vibration isolation materials and mounts as a function of their operating conditions and certain dynamic properties. We shall relate the load on the mount, the shape of the mount and the frequency function of the modulus of elasticity of the material to these conditions and properties.

We transform formula (181), substituting the expression for the natural frequency of the machine as mounted on vibration isolation mounts:

$$VI = 40 \log \frac{f}{\frac{1}{2\pi}\sqrt{\frac{C}{m}}} = 40 \log \frac{f}{\frac{1}{2\pi}\sqrt{\frac{ESg}{hG}}},$$

where G is the weight of the machine;

g is the acceleration of the force of gravity;

the other notations have been introduced earlier.

Taking into account the fact that $\frac{G}{S} = p_0$ is the specific pressure, or loading on the mount, and separating out the quantities of interest to us, we obtain:

$$VI = 20 \log \frac{p_0}{E} + \text{const db.} \qquad (182)$$

It is apparent that the vibration isolation increases with an increase in the loading on the mount, although only up to a certain limit, which for rubber may be taken as equal to 8 or 10 kG/cm^2, and for cork 2 kG/cm^2. When the load is increased above this limit considerable internal stresses are engendered within the mount, leading to an increase in its effective modulus of elasticity and this, as may be seen from formula (182), reduces the vibration isolation. Rubber, furthermore, begins to "flow" at such high loadings, i.e. irreversible deformation takes place, causing a general settling down of the machine on the mounts.

The matter of their shape is directly related to the problem of the effect of the loading on rubber mounts. Rubber is one of the practically incompressible materials (coefficient of Poisson 0.48 - 0.49), and its deformation is due exclusively to a change in shape, and not in volume. If the mount is bounded on its sides by a metal fitting, or if it has a large ratio of supporting surface to side surface, it will conduct vibration very well because it is unable to expand toward the side.

In Example 5 it was found that in wide rubber mounts, the speed of sound is several-fold greater than its speed in narrow mounts. Inserting the value of the speed of sound

$$c = \sqrt{\frac{E}{\rho}}$$

into formula (182), we obtain:

$$VI = 40 \log \frac{1}{c} + \text{const db.} \qquad (183)$$

Taking into account the earlier findings with respect to the magnitude of the speed of sound in wide mounts, we see that their vibration isolation capacity is less than that of narrow mounts. To eliminate this undesirable characteristic in wide mounts, they must be divided by as many cuts, openings and sections as possible. Experiments performed by the present author indicate that the vibration isolation of a rubber mount 100 x 200 x 20 mm is 10 to 12 db lower in

a frequency range of 100 to 700 cycles than that of the same mount cut into 25 parts. Mounts consisting of a series of staggered and glued together monolithic and perforated rubber layers have very good vibration isolation properties. The air cavities due to the perforations enable those portions of the rubber which are remote from the free surfaces of the mount to expand laterally under vibration.

With respect to the dependence of the modulus of elasticity of rubber (or, correspondingly, the speed of sound in it) upon frequency, according to a great deal of data of Soviet and foreign investigators this dependence may be represented in the following form for a broad range of frequencies:

$$E = E_{st}\left(1 + af^{n}\right), \tag{184}$$

where a and n are positive numbers.

Substituting this expression into formula (182), we obtain:

$$VI = 20 \log \frac{1}{E_{st}(1 + af^{n})} + \text{const db.} \tag{185}$$

The greater the increase of the modulus with frequency, the more marked is the drop in vibration isolation at high frequencies. There is reason to believe that rubber is characterized by a relatively weak dependence of the modulus of elasticity upon frequency, especially in the case of styrol rubber. This type of rubber must be used in vibration isolation mounts.

47. The Effect of the Foundation Characteristics on the Vibration Isolation of a Vibration Mount

The properties of the foundation (its mass and stiffness) exert an influence on the acoustic effect of vibration mounts. Because under shipboard conditions light foundations often are used, their effect should be evaluated, if only approximately. It may be mentioned that, as between the true vibration isolation of the mounts and approximate value thereof as determined by practical measurements both in ships and on regular anti-vibration installations, this effect is different.

Let us imagine a machine mounted on vibration mounts attached to a foundation of finite mass and stiffness, in a system with two degrees of freedom (Figure 142-a). In contrast to a foundation of infinite mass, here it is possible and advantageous to operate upon the values of the vibratory amplitudes or the vibratory velocities of the machine and the foundation. The value of the true isolation of the mounts at a given frequency is determined by the ratio of the vibratory

velocities or amplitudes of a fixed point of the foundation when it is rigidly and when it is elastically attached to the machine:

$$VI' = \frac{\dot{a}_{1zh}}{\dot{a}_1} = \frac{a_{1zh}}{a_1}. \tag{186}$$

Here the index $_1$ relates to the foundation, and the index $_{zh}$ indicates rigid attachment of the machine to the foundation (Figure 142-b).

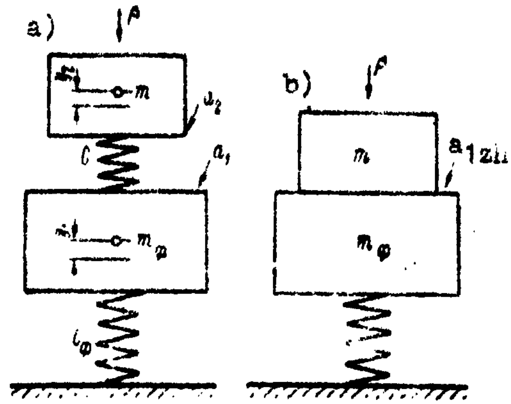


Figure 142. Defining the concept of the vibration isolation of vibration mounts and the drop in the levels of sonic vibration in mounts set on a foundation of finite mass.

Because in practice it is very difficult to measure vibration levels of a foundation first with rigid, and then with elastic attachment of the machine, we usually limit ourselves to measuring the drop in sound amplitude at the vibration mount. In the symbols of Figure 142-a, this drop is:

$$D' = \frac{\dot{a}_2}{\dot{a}_1} = \frac{a_2}{a_1}. \tag{187}$$

It is necessary to determine the relationship between the value of the vibration isolation of the mounts and the parameters of the foundation and the machine, and also the relation of the isolation to the value of the drop in sound levels at the shock absorbers.

Let us write a system of equations for the motion of the two-mass system of Figure 142-a according to the type of equation (11), assuming that friction is absent.

$$
\left.\begin{aligned}
m\ddot{y}_2 + C(y_2 - y_1) &= F_a e^{j\omega t};\\
m_\Phi \ddot{y}_1 + (C_\Phi + C) y_1 - C y_2 &= 0.
\end{aligned}\right\} \tag{188}
$$

Here m, m_Φ are the masses of the machine and the foundation;

C and C_Φ are the total stiffness of the mounts and a certain "adduced" stiffness of the foundation;

F_a is the amplitude of the vibratory force F.

Taking the value of the amplitude of vibration of the foundation and the machine in the form:

$$
y_1 = a_1 e^{j\omega t}, \quad y_2 = a_2 e^{j\omega t}, \tag{189}
$$

we obtain from the preceding system of equations:

$$
\left.\begin{aligned}
-Ca_1 + (-m\omega^2 + C) a_2 &= F_a,\\
(-m_\Phi \omega^2 + C_\Phi + C) a_1 - C a_2 &= 0,
\end{aligned}\right\} \tag{190}
$$

therefore the amplitude of vibration of the foundation is:

$$
a_1 = \frac{F_a}{\left(1 + \frac{C_\Phi}{C} - \frac{\omega^2 m_\Phi}{C}\right)(C - m\omega^2) - C}.
$$

The same, with the machine fastened rigidly to it ($C = \infty$):

$$
a_{1\mathrm{zh}} = \frac{F_a}{C_\Phi - (m + m_\Phi)\omega^2}.
$$

The vibration isolation of the mounts in relation to vertical vibration:

$$
\mathrm{VI}' = \frac{a_{1\mathrm{zh}}}{a_1} = \frac{\left(1 + \frac{C_\Phi}{C} - \frac{\omega^2 m_\Phi}{C}\right)(C - m\omega^2) - C}{C_\Phi - (m + m_\Phi)\omega^2}. \tag{191}
$$

At frequencies 3- to 4-fold greater than the natural frequencies of the foundation and of the machine as mounted, the elastic resistances may be ignored in comparison to the inertial resistances.

From equation (191) we obtain (dropping the sign indicating the phase of the vibrations):

$$\mathrm{VI}' = \frac{\omega^2 m_\phi m}{C(m_\phi + m)} = \left(\frac{\omega}{\omega_0}\right)^2 \cdot \frac{m_\phi}{m_\phi + m}.$$

But as was seen from formulas (180) and (181), $\left(\frac{\omega}{\omega_0}\right)^2$ at these frequencies is the isolation of the mounts with a foundation of infinite mass - VI'_∞, i.e.

$$\frac{\mathrm{VI}'}{\mathrm{VI}'_\infty} = \frac{m_\phi}{m_\phi + m}.$$

The difference in the values of vibration isolation, in decibels, with a finite, and with an infinitely large mass of the foundation is:

$$\Delta \mathrm{VI} = \mathrm{VI} - \mathrm{VI}_\infty = 20 \log \frac{\mathrm{VI}'}{\mathrm{VI}'_\infty} = 20 \log \frac{\frac{m_\phi}{m}}{\frac{m_\phi}{m} + 1} \text{ db.} \qquad (192)$$

This function is plotted graphically in Figure 143 (curve 1). As is apparent, the value of $\Delta \mathrm{VI}$ is negative, i.e. the isolation of the mounts installed on a foundation of finite mass always is less than the isolation of the same mounts when set upon an extremely massive foundation. At $|\Delta \mathrm{VI}| > \mathrm{VI}_\infty$, which is characteristic of light foundations, the true isolation is negative, i.e. the vibration of the foundation is increased in comparison to the case of rigid attachment of the machine.

In similar manner we determine the value of the drop D'. From the second equation of system (190) it follows that:

$$D' = \frac{a_2}{a_1} = 1 + \frac{C_\phi}{C} - \frac{\omega^2 m_\phi}{C}. \qquad (193)$$

Inserting this into equation (191), we obtain for the determined frequencies of vibration:

$$\mathrm{VI}' = D' \frac{m}{m + m_\phi}.$$

The difference between the true, and the measured vibration isolation, in decibels, is:

$$\mathrm{VI} - \mathrm{D} = 20 \log \frac{\mathrm{VI}'}{\mathrm{D}'} = 20 \log \frac{1}{1 + \frac{m_\Phi}{m}}. \qquad (194)$$

The corresponding curve 2 in Figure 143 indicates that this difference is negative, i.e. that the drop in sonic vibration at the mounts always is greater than their true vibration isolation.

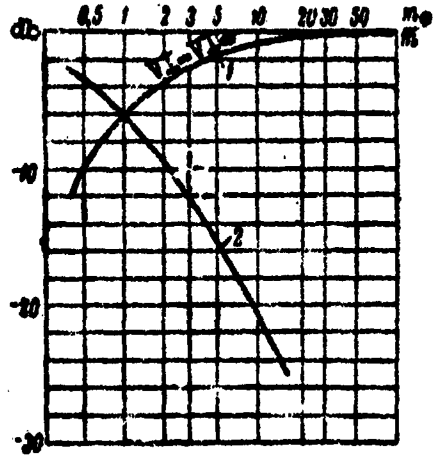


Figure 143. Effect of the mass of the foundation upon the vibration isolation of vibration mounts.
1 - Difference in the values of vibration isolation with finite, and infinitely large mass of the foundation;
2 - Difference between the true value of vibration isolation of the mounts and the value of vibration isolation measured under actual conditions (drop in the levels of sonic vibration at the mounts).

The above conclusion holds for frequencies up to the point at which the machine and the foundation may be considered as moving as a whole. However, even at higher frequencies the same expressions for ΔVI and VI-D hold true if the machine and the foundation are considered as objects having a uniform distribution of sound energy.

Example 48. A machine mounted on vibration mounts, originally installed on a very massive foundation $\left(\frac{m_\Phi}{m} \approx 5\right)$, was transferred to a light deck $\left(\frac{m_\Phi}{m} \approx 0{,}5\right)$. The problem is to determine the change in the acoustic effect of the vibration isolation of the machine.

Solution. From curve 1 of Figure 143, for $\frac{m_\phi}{m}=5$, $VI_1 = VI_\infty - 2$ db; and from the same at $\frac{m_\phi}{m}=0{,}5$, $VI_2 = VI_\infty - 10$ db. Therefore, with transferral of the machine on its mounts to a light foundation, the vibration isolation decreases by the amount

$$VI_1 - VI_2 = 8 \text{ db}.$$

If in this case the mounts were such that $VI_\infty < 10$ db, then the vibration of the light foundation will become greater if the machine is installed on the mounts.

The stiffness of the plates of the foundation also is very important from the point of view of achieving the acoustic effect of vibration isolation treatments.

For the purpose of increasing the vibration isolation of mounts, W. Kuhl (143) suggested that supplementary masses (Figure 144) be installed beneath the mounts in the case of relatively compliant foundations.

According to the graphical data of Kuhl, the gain in the value of vibration isolation for the case of installation of supplementary masses M_{sup} under each mount attains the following values:

	Increase in vibration isolation, db			
	$\frac{f}{f_0}$	2	5	10
M_{sup}	0.5 M'	6	8	10
	M'	9	11	14

Here f is the frequency at which the increment in vibration isolation was determined;

f_0 is the frequency of free vibration of the whole installation with mounts;

M' is a quantity having the dimension of mass, and equal to:

$$M' = m_p \sqrt[3]{\frac{EI}{C}}, \tag{195}$$

where m_p is the weight per unit length of a foundation girder;

E and I are the modulus of elasticity of the girder material and the moment of inertia of its cross-section, relative to the horizontal axis of inertia;

C is the vertical stiffness of the vibration mount.

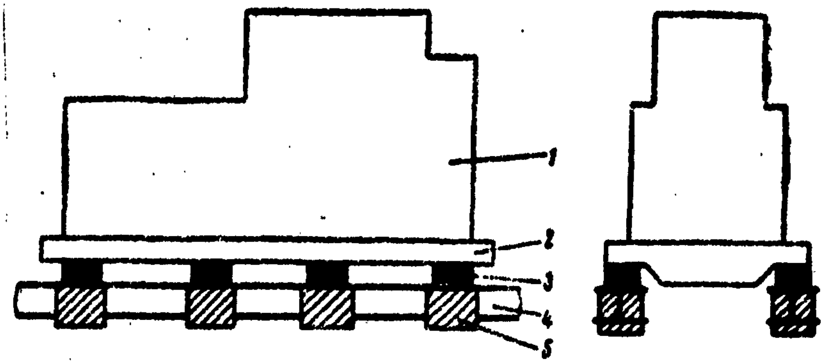


Figure 144. Supplementary masses installed under vibration mounts for the purpose of increasing their vibration isolation. 1 - Machine; 2 - Bed plate of the machine; 3 - Vibration isolation mounts; 4 - Foundation girder; 5 - Supplementary masses.

Example 49. A machine mounted on vibration isolation mounts is installed on a light deck, on a foundation consisting of two parallel angle bars 110 x 70 x 6.5 mm, with unequal sides, the larger of which is horizontal. The stiffness of each mount is 200 kG/cm. The problem is to determine the supplementary mass to be placed under each mount in order that their vibration isolation will be increased by 11 to 13 db at a frequency of $f > 5 f_0$.

Solution. From the steel handbook we find the moment of inertia for profile No. 11/7 relative to the axis parallel to the larger side of angle bar, to be $I \cong 74$ cm^4, and the linear weight of this bar is $m_p = 11.4$ kg/m. Inserting these data, and also data on the stiffness of the mounts, in formula (195), we obtain:

$$M = m_p \sqrt[3]{\frac{EI}{C}} = 11{,}4 \cdot 10^{-2} \sqrt[3]{\frac{2 \cdot 10^6 \cdot 74}{200}} \cong 10 \text{ kg}.$$

If a mass of 5 kg is placed under each mount, the vibration isolation at a frequency of $f > 5f_0$ will increase by 8 to 10 db.

48. Vibration Inhibiting Masses

An elastic wave propagating through a metal structure may be arrested not only by a vibration isolating insert, but also by some sort of barrier having an impedance considerably greater than the impedance of the structure. L. M. Brekhovskikh (25) mentioned the possibility of the use of "massive" barriers for this purpose. Later, L. Cremer (124) indicated that the vibration isolating effect of a barrier of this type depends not only upon the absolute magnitude of the mass, but also the magnitude of the moment of inertia of the barrier relative to an axis passing through the place at which it is fastened to the vibrating surface. The greatest value of moment of inertia for a given mass is possessed by a body in the form of a wedge or a "T" beam, standing on edge (Figure 145), which for this reason are suitable for use as vibration inhibiting masses.

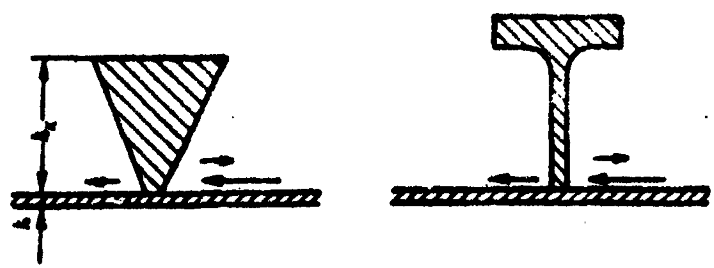


Figure 145. Vibration inhibiting masses serving to isolate flexural waves propagating through a plate.

A mass with radius of inertia r isolates the flexural vibrations of a plate, the wave length of which satisfies the condition:

$$\lambda_{fl} = \frac{c_{fl}}{f_{fl}} < 2\pi r, \tag{196}$$

where f_{fl} is the frequency at which the vibration isolation effect of the mass appears;

c_{fl} is the speed of propagation of flexural waves in the plate.

Substituting here the value of c_{fl} as obtained from formula (41), after simple transformations we obtain:

$$r = 150\sqrt{\frac{h}{f_{fl}}}, \tag{197}$$

where h is the thickness of the plate, cm.

It may be indicated that in the case of a wedge, the basic radius of inertia r_w relative to an axis passing through its vertex and parallel to its base, regardless of its height and width proportions, is:

$$r_w = \frac{h_w}{\sqrt{2}} .$$

Inserting this expression into formula (197), we obtain the following value of the necessary height of the vibration-inhibiting wedge:

$$h_w = 210\sqrt{\frac{h}{f_{fl}}} . \tag{198}$$

At the linear weights of vibration inhibiting masses permissible in practice, their vibration isolating effect at frequencies above f_{fl} attains 10 or 12 db. At very high sonic frequencies the isolation of flexural vibrations again drops.

In contrast to elastic inserts, masses provide poor isolation of longitudinal vibrations of all frequencies in plates.

The use of several identical vibration inhibiting masses placed in series one after another does not markedly increase the vibration isolation effect of the first of those masses, and therefore such a system of vibration isolation, which would also entail a considerable increase in weight, may hardly be recommended for use.

Figure 146 shows an example of the installation of a vibration inhibiting mass on the shell of a ship for the purpose of reducing the transmission of sonic vibration from a machine to the walls of an adjacent quiet room.

The frames of a ship also have a certain degree of vibration isolating action. Their vibration isolating effect (in a direction perpendicular to the frames) attains 1 to 3 db at medium sonic frequencies. It may be increased somewhat by welding massive plates to that part of the frame most remote from the shell plating, thereby increasing the moment of inertia of the cross-section of the frame.

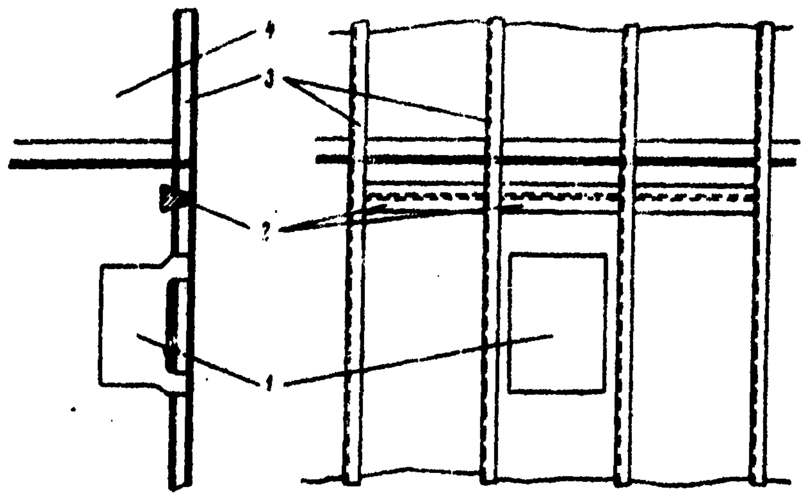


Figure 146. Installation of vibration inhibiting wedges on the shell of a ship.
1 - Source of sonic vibration; 2 - Vibration inhibiting wedge; 3 - Frames; 4 - Sound-isolated compartment.

Vibration isolating structures are satisfactorily effective only with the simultaneous application of a vibration absorbing treatment to the plates (cf. Paragraph 54). The use of vibration isolating masses, alone, leads to a concentration of sound, which is reflected by them in the same manner in which sound is reflected under a sound-protective housing which lacks an internal sound-absorbing coating, and therefore isolates sound poorly.

Example 50. The problem is to determine the linear weight of a vibration inhibiting mass in the form of a wedge, installed on a 5-mm-thick shell plate. The vibration isolation effect of the mass must appear at frequencies above 200 cycles.

Solution. Inserting the initial data in formula (198), we obtain the necessary height of the wedge:

$$h_w = 210\sqrt{\frac{0.5}{200}} = 10.5 \text{ cm}.$$

If the width of the base of the wedge is one-half of its height, the linear weight of the steel wedge is equal to:

$$G = \left(\frac{1}{2}\,1{,}05\right)^2 \cdot 7{,}85 \cong 21{,}5 \text{ kg/m}.$$

CHAPTER 13

THE DESIGN AND ANALYSIS OF SOUND AND VIBRATION

ISOLATING DEVICES

49. Designs for Vibration Isolation Mounts

Figure 147 shows several designs of welded rubber-metal vibration isolation mounts, i.e. mounts in which the rubber element is joined ("vulcanized") to metallic retaining elements.

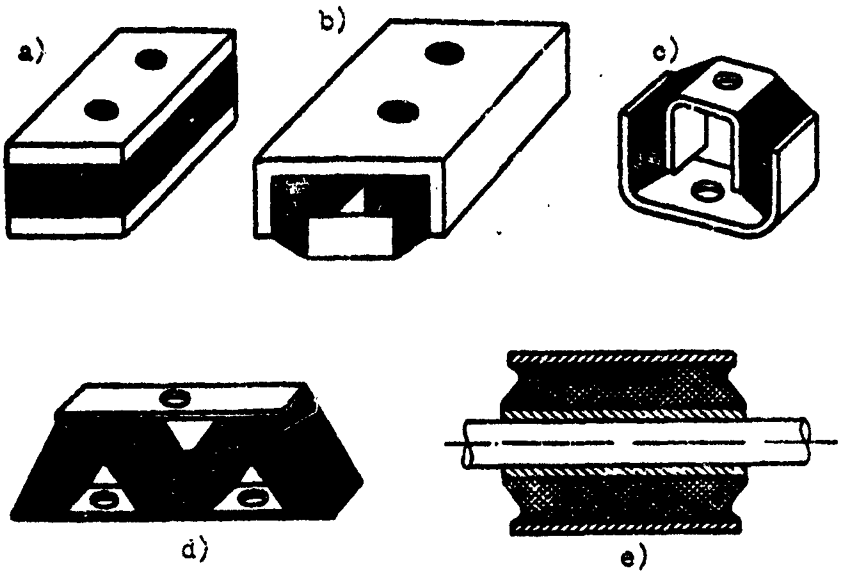


Figure 147. Designs of vulcanized rubber-metal vibration isolation mounts [rezinometallicheskie amortizatory].

The simplest design consists of a two-plate vulcanized mount (Figure 147-a). The upper and lower metal plates of the mount may have various planar shapes, such as rectangular, round, or hexagonal. The mount is fastened to the foundation and to the bed plate or a leg

of the machine with bolts or pins, screwed into the retaining metal plates or welded to them.

A not very desirable property of these simple mounts is the great difference between their stiffness in the axial and in the transverse directions. In small two-plate vibration mounts the ratio of the stiffness in compression and in shear (transversely) is 5 or 6, and in mounts with a large supporting area, in which the rubber insert is enclosed to a large extent by metal plates, this ratio attains 15 to 20. To reduce the difference in stiffnesses somewhat, rubber is inserted into the mounts in such a way as to act in compression transversely (Figure 147-b), which complicates the design of the mount.

The mount shown in Figure 147-c in which the rubber element in the axial direction functions in shear, and the very soft mount, shown in Figure 147-d, are used mainly for cushioning instruments, although they may be applied for vibration isolation of light weight machines. A mount of the "sleeve" type, (Figure 147-e) is used for suspending machines from the overhead or attaching them to bulkhead walls.

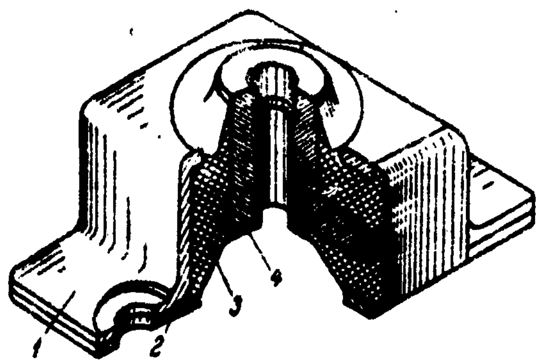


Figure 148. AKSS vibration isolation mount.
1 - External bracket with openings for fastening to the foundation; 2 - Lower plate; 3 - Rubber mass; 4 - Inner bushing with threaded opening for fastening to bed plate or leg of a machine.

All the mounts shown in Figure 147, with the exception of the "sleeve" type, have the disadvantage that if the rubber element becomes detached from the metal plates (in the course of time or due to accidental jarring), the machine may tear loose from the mounts. This shortcoming is eliminated by the AKSS type mount developed by the present author, a vulcanized shipboard vibration isolation mount "with insurance" (Figure 148). In this case the design of the metal fittings is such that damage at the places where the rubber is joined to the

metal does not lead to disintegration of the mount. The AKSS mounts have found extensive application not only on various types of ships, but also in industrial and civil construction.

The original AKSS mounts were produced by Soviet industry with soft, non-oil-resistant rubber. Later, hard, oil-resistant rubber began to be used (AKSS-M type mounts), but this lowered their sound isolating properties. In recent years the production of shock absorbers with soft, oil-resistant rubber was mastered, and thus at the present time mounts of the type AKSS-I have a sufficiently high sound isolation effect, plus resistance to oil and diesel fuel. The frequency of free vertical vibration of machinery on AKSS mounts, at the nominal loading, ranges from 10 to 15 cycles; deformation at nominal loading is 1 to 1.5 mm.

Table 23

Vulcanized Rubber-Metal Vibration Isolation Mounts, Type AKSS-I

Type-Size Number	Name of Mount	Nominal Loading on Mount, kg	Vertical Dynamic Stiffness, kG/cm
1	AKSS-25I	25	500
2	AKSS-40I	40	650
3	AKSS-60I	60	1000
4	AKSS-85I	85	1350
5	AKSS-120I	120	1200
6	AKSS-160I	160	2600
7	AKSS-220I	220	4000
8	AKSS-300I	300	4000
9	AKSS-400I	400	5300

Table 23 gives the type-sizes of AKSS mounts plus their stiffness and loading. The design of AKSS mounts developed later for loads in excess of 160 kg differ slightly from that shown in Figure 148. The rubber mass in those mounts is enclosed to a great extent by the metal bracket, which reduces their sound isolation somewhat in comparison to the smaller mounts.

More complex designs of shipboard vibration mounts, type APR, with a combination of elastic elements (rubber and spring), type APS with a pneumatic element, vulcanized rubber-metal mounts with a very tall isolating element for 1 to 2-ton loads, etc., also are produced by Soviet industry.

When it is not possible to install vulcanized rubber-metal mounts, built-up mounts with elastic inserts cut from thick rubber bars or consisting of several layers of sheet rubber, preferably perforated, glued together, may be employed. Mounts of this type (Figure 149), with inserts of regular rubber sheet were applied successfully for mounting the engines of river passenger launches (48). In built-up mounts, as in the mounts shown in Figure 147-b, rubber elements acting laterally in compression are provided for the purpose of increasing the stiffness in a transverse direction.

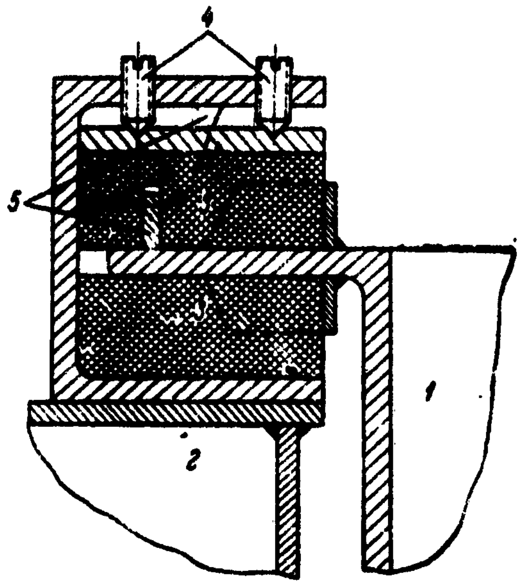


Figure 149. Built-up rubber mount.
1 - Floating bed-plate of the machine; 2 - Plate of the foundation; 3 - Rubber inserts; 4 - Screws regulating compression in the inserts; 5 - Brackets for increasing transverse stiffness.

In the designing and installing of built-up mounts it is equally important to ensure the absence of excessive compression of the elastic inserts by the fastening fitting, and to ensure a desirable degree of separation of the sound-isolated metal parts, because even a small metallic "bridge" greatly reduces the sound isolation of vibration mounts at high frequencies.

Figure 150 shows a mount made of steel strip springs, which also may be applied successfully on shipboard. To reduce transmission of vibration at wave resonance frequencies a vibration absorbing (left)

or vibration isolating (right) element of rubber or plastic composition is introduced.

Elements having internal friction, such as rubber inserts, may be replaced in metal mounts by elements with surface friction. Thus according to foreign data (141), mounts consisting of an assemblage of steel springs have good sound isolation.

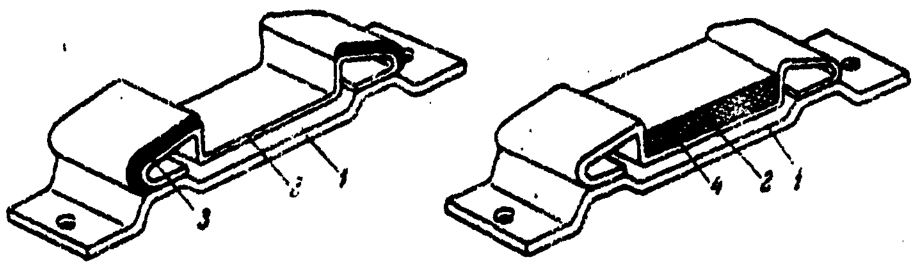


Figure 150. Vibration isolation mount of strip spring steel for shipboard machinery.
1 - Supporting bracket; 2 - Elastic element; 3 - Vibration absorbing layer; 4 - Sound isolating insert of elastic material with high coefficient of mechanical losses.

When cylindrical springs are used in isolation mounts, it is advantageous to insert rubber or felt bushings under the springs for better isolation of high frequency vibrations. The combination of these bushings with steel springs developed by us permits very great loadings, and as a result, reduces the frequency of free vibrations of the machine as mounted, and enables attainment of a very considerable acoustic effect in a broad range of frequencies.

50. Calculation of the Frequency of Free Vibrations and the Magnitude of the Displacement of Machines Mounted on Vibration Isolation Mounts

Vibration isolation mounts for shipboard machinery may be divided into two types for purposes of their analysis:

1) those for well-balanced machines, not subjected to the underway vibration of the ship;

2) those for machinery subject to internal or external periodic exciting forces.

Installations having mounts of the first type require no special vibration computation. For those machines, the calculation usually is limited to determining the frequency of free vertical vibration of the machine on the mounts, which is a kind of criterion for the degree of vibration isolation (Paragraph 45). In the case of non-balanced machines mounted on vibration isolation mounts and machines installed at the location of intense vibration, all frequencies of free vibration must be calculated, and in some cases also the amplitude of forced vibrations also must be determined.

The expression for free vertical vibrations of an item of machinery mounted on isolation mounts has been introduced earlier:

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{C}{M}}, \tag{199}$$

where C is the total vertical stiffness of the mounts;

$M = \frac{G}{g}$ is the mass of the machine (G is the weight of the machine, and g is the acceleration due to gravity).

Taking account of the fact that $\frac{G}{C} = \delta_{cm}$ is identical with the static flection of the mounts, in centimeters, after simple transformation of formula (199) we find:

$$\delta \cong \frac{250}{f_0^2}, \tag{200}$$

where δ is the deflection in millimeters.

A simple nomogram (Figure 151) may be adduced for the rapid determination of the frequency of free vertical vibration and the static deflection of objects mounted on vibration mounts. The entering and sought values vary on the nomogram within wide limits. However, assuming adequate vibration isolation, the value of the main desired quantity f_0 must be limited to 15 cycles (striated area on the right-hand scale).

Example 51. A machine weighing 215 kg is mounted on 4 isolation mounts with 200 kG/cm stiffness, each. The problem is to determine the frequency of free vertical vibration and the static deflection of the mounts.

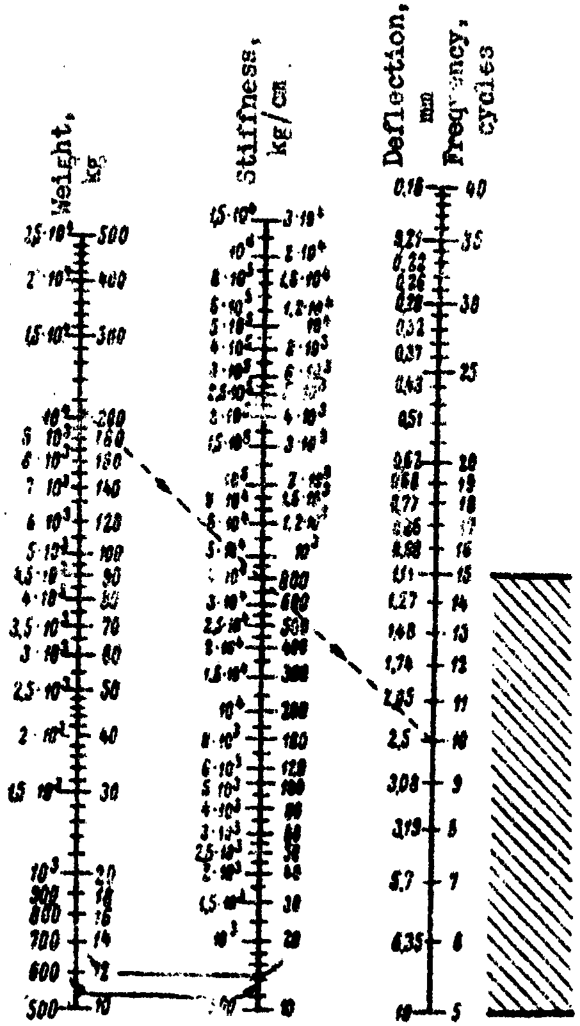


Figure 151. Nomogram for determining the frequency of free vibration of a machine mounted on isolation mounts and the static deflection of the mounts.
The striated area shows the recommended frequencies of free vertical vibration for use in ships.

Solution. Joining the original values G and C on the nomogram of Figure 151 with a straight line, we read at the point of intersection with the right-hand scale: f_0 = 10 cycles; δ = 2.5 mm.

It is apparent (cf. left portions of the scales of weight and stiffness), that these values of f_0 and δ will apply also in the case of a machine 10.5 tons in weight, having mounts with a total stiffness of 40,000 kG/cm.

The values of stiffness required for the calculations are supplied by the manufacturer of the mounts. In developing new designs for ordinary rubber mounts, or in evaluation of the calculated stiffness of them, the following simple method may be used, which takes account of the non-linearity of the elasticity of rubber. The latter has a marked effect when the rubber is bounded within the sides of the mount.

The stiffness of a rubber insert is presented in the form:

$$C = k\frac{ES}{h}, \tag{201}$$

where S, h and E are the supporting area, height and modulus of elasticity of the insert material, respectively;

k is a coefficient, which may be called the containment stiffness coefficient. The value of this coefficient is greater, the fewer the free surfaces at the boundaries of the material (including the supporting surfaces). It also depends upon the specific pressure on the material, p_0.

Figure 152 presents a graph of k as a function of p_0 and k_{pl} which was obtained experimentally by the author, giving the ratio of free boundary area of the elastic material to the total boundary area, expressed in %. Semi-hard rubber materials ($E = 40$ to 50 kG/cm^2), with a square and round form to their bases, were used in the experiment; however, the functional relationship obtained also holds approximately for rubber of other shapes, and consisting of softer or harder material.

The striated area of the graph relates to large values of k and to small values of p_0, which are undesirable from the point of view of vibration isolation (cf. formula 182). With substantial free side area of the bushing ($k_{pl} = 50$ to 66 %), increasing the load on the material changes its stiffness relatively little. It may be noted that $k_{pl} = 66\%$ corresponds to a cube, bounded only on the load-carrying surfaces.

<u>Example 52</u>. The problem is to find the amount of increase in the stiffness of a rubber element 20 x 10 x 4 cm mounted under a

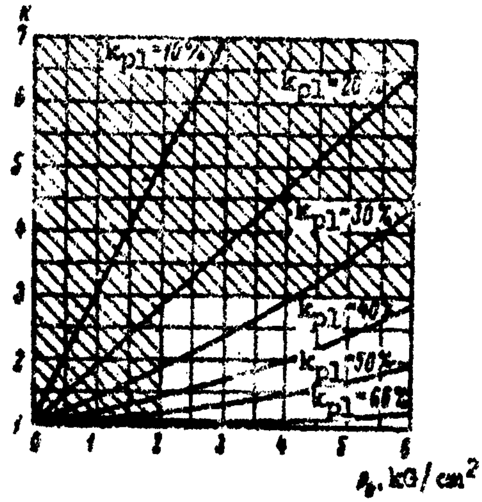


Figure 152. Coefficient of stiffness of a rubber element as a function of the coefficient of free boundary area and specific pressure. Striated area indicates undesirable values of k_{pl} and p_0.

marine diesel engine, when its height is reduced two-fold. The loading on the element is 4.5 kG/cm^2.

<u>Solution</u>. The coefficient of free area of the element prior to reduction of its height, is:

$$k_{pl_1} = \frac{2\cdot 4\,(20+10)\times 100}{2(20\cdot 10+20\cdot 4+10\cdot 4)} \cong 37\%.$$

According to the graph of Figure 152, at $p_0 = 4.5$ kG/cm^2, $k_1 \cong 2.5$.

For an element of two-fold less height we have:

$$k_{pl_2} = \frac{2\cdot 2\,(20+10)\times 100}{2(20\cdot 10+20\cdot 2+10\cdot 2)} = 23\%.$$

and therefore $k_2 \cong 4.5$.

The ratio of stiffnesses of the elements (formula 201) is:

$$\frac{C_2}{C_1} = 2\,\frac{4{,}5}{2{,}5} = 3{,}6.$$

In the isolation of machines subjected to vibration we must take into account, as mentioned previously, all the degrees of freedom of the machine, which in the general case comprise six. They correspond to linear vibrations of the mechanism in the directions of the three coordinate axes, plus rotatory vibration around these axes.

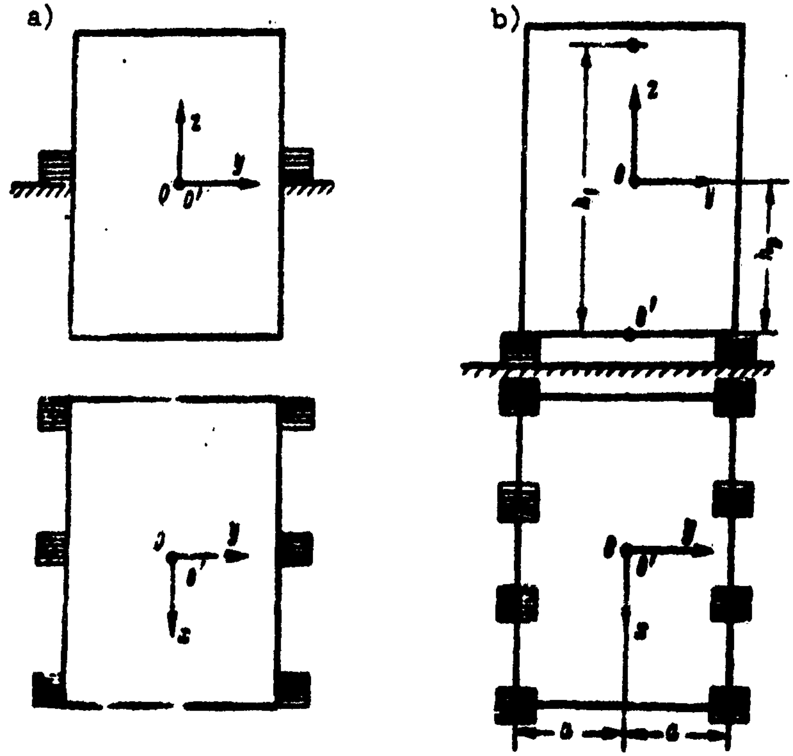


Figure 153. The most important cases of the arrangement of elastic supports beneath a machine.

Two important special cases of the arrangement of elastic supports and of the calculation of the frequency of free vibrations of machinery mounted on isolation mounts must be mentioned.

In the first case the center of rigidity of the elastic base coincides with the center of inertia, due to high placement of the supports of the machine (Figure 153-a). This case is to be the goal at all times. Here, all six frequencies of free vibration may be found through simple, and independent solutions. This fact means that resonance of any given type of vibration does not evoke simultaneous resonant vibration of another type. To avoid intense vibration of a

machine mounted on isolation mounts, the frequencies of free vibrations must be outside the zone of frequencies of the most intense exciting forces at the principal operating conditions of the machine.

In another, more frequently encountered case (Figure 153-b), the center of rigidity of the elastic base is located on a vertical line passing through the center of inertia of the body. In this case the frequency of transverse and of rotational vibration in the planes zoy and zox are coupled together (so-called double-coupled vibration), and the frequency of free vertical vibration and of rotational vibration relative to the vertical axis are found by independent solutions. Many authors, such as (122), have proposed nomograms for simplification of the calculation of frequencies in this basic case of vibration isolation mounting.

If the projection of the center of inertia onto the plane of the mounts does not fall on their center of stiffness but lies on one of the axes of the elastic base, the expressions for the frequencies of free vibrations are triple-coupled. In a still more complex case, when the projection of the center of inertia coincides with none of the principal axes of the elastic base, the expressions for all six frequencies of free vibration are inter-connected. In both of these cases the probability of appearance of intense vibration in unbalanced machines increases, because linear vibrations arise not only from an exciting force, but also from the exciting moment of the same frequency, and rotational vibrations arise not only from an exciting moment, but from an exciting force, as well. For this reason, and because of the complexity of performing vibration computations, arrangements of vibration mounts characterized by triple-, and sextuple coupled vibrations must be avoided.

Vibration isolation arrangements for shipboard machinery having external shaft-, or pipe connections must be checked with respect to the magnitude of the inclination deflection of the machine during rolling of the ship, in addition to the usual vibration check. This value must not exceed 2 to 3 mm at the points of attachment of external supporting connections of the machine at maximum roll angles (50 to 55°). The magnitude of the deflection of a machine on vibration mounts, in a transverse direction, is determined in general form by inertial forces and the components of the force of weight during inclination of the machine. Excluding cases in which the machine is greatly removed from the center of roll, the forces of inertia may be ignored. In this case the inclination deflection of points of a machine located at distance h_1 from the plane of support, is:

$$\Delta_{\kappa} \cong G \sin \varphi_{\kappa} \left(\frac{1}{C_y} + \frac{h_0 h_1}{C_\varphi a^2} \right), \tag{202}$$

where φ_K is the angle of roll;

C_z and C_y are the total vertical and transverse stiffnesses of the mounting;

The other notations are explained in Figure 153-b.

When a machine is mounted on an elastic base, the supplementary stiffness obtained by virtue of the foundation is eliminated, and the strength of the body of the machine may be inadequate. Because of this the body of the machine must be checked for strength. If this strength is inadequate, the machine is to be mounted on a rigid frame, under which the vibration mounts are placed. The introduction of a common frame is mandatory in providing vibration isolation for machinery installations consisting of two or three units.

51. Vibration Isolation With Inclined Supports

The usual method of vibration isolation mounting with vertical supports has at least two substantial shortcomings. The first, as had been seen in the foregoing, is related to the rotational and transverse vibrations which arise when the center of gravity is located outside the plane of the supports, which almost always is the case. The interconnection of vibrations complicates the numerous practical computations for vibration isolation, and complicates the selection of frequencies for isolated machinery, because changing the parameters of the elastic base for the purpose of changing one of the frequencies of interconnected vibrations does not always lead to a desirable change in another frequency.

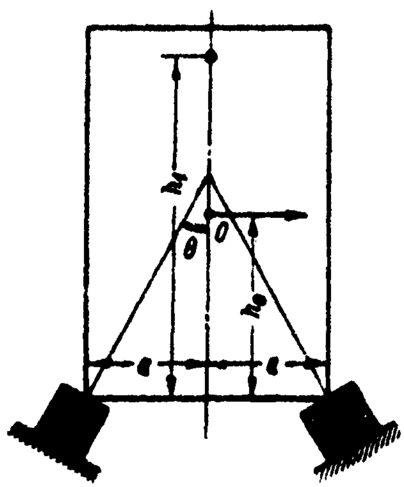


Fig. 154. Vibration isolation mounting with inclined supports.

Another shortcoming, and one which is perhaps more important with respect to shipboard conditions, of vibration isolation with vertical supports is its relatively low transverse stiffness, which is accompanied by a considerable transverse displacement of the machine during rolling of the ship, bumping against piers and other types of shocks, and thus the possibility of damage to the piping, shafting and cables connected to the machine.

Transverse displacement of machinery may be reduced by special elastic side supports or vibration isolation mounts with elastic bounding surfaces. The desired result often

may be attained by inclining both rows of supports inwards (Figure 154). A similar type of mounting is that used in setting aircraft engines in a circular frame.

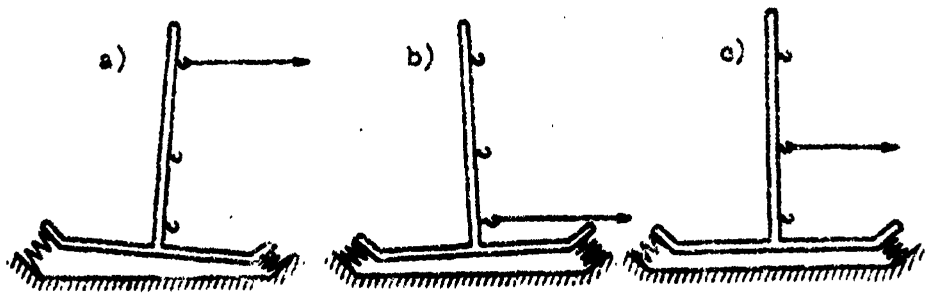


Figure 155. Model illustrating the character of the displacements of a body mounted on inclined supports under the effect of a transverse force: a - force applied high; in addition to displacement toward the right, the body also is rotated toward the right; b - force applied low; in addition to displacement toward the right, the body rotates toward the left; c - force applied at a certain fictitious center of stiffness of the elastic base with inclined supports evokes only a displacement of the body in the direction of the action of the force.

Figure 155 clearly shows why the application of a horizontal force at the center of stiffness in the case of a particularly inclined support results in only horizontal displacement of the machine, not accompanied by rotation. In distinction from the case illustrated in Figure 153-b, transverse and rotatory vibrations have been separated, making possible selection of the frequencies of these vibrations independently of each other.

We may convert expression (202) for the transverse displacement of the higher points of a machine mounted on isolation mounts, and located distance h_1 from the plane of the supports, under the effect of a horizontal force, such as that arising during the rolling of a ship:

$$\Delta_{\kappa} = G \sin \varphi_{\kappa} \left(\frac{1}{C_y} + \frac{h_0 h_1}{C_\varphi a^2} \right). \tag{203}$$

With inclined supports, the second member of the expression, which is due to rotation of the machine on the mounts, disappears.

The value C_y in the first member changes, and becomes equal to (49):

$$C_y = \left(\sin^2\theta + \frac{1}{K}\cos^2\theta\right) C_1, \tag{204}$$

where $K = \frac{C_1}{C_2}$; C_1 and C_2 are the stiffnesses of the mounting in the direction of the two principal elastic axes of each of the mounts;

θ is the angle of inclination of the mounts to the z axis.

The ratio of the transverse displacements of points of a machine with ordinary and inclined mounts, from formulas (203) and (204) is:

$$\frac{\Delta_K}{\Delta_{Kz}} = \left(\sin^2\theta + \frac{1}{K}\cos^2\theta\right)\left(K + \frac{h_0 h_1}{a^2}\right). \tag{205}$$

The value of K for simple two-layer isolation mounts, as were discussed earlier, varies from 5 or 6 for small-size mounts to 15 or 20 for large ones. Because of this, for mount angles of $\theta > \sim 30°$, the second member within the first parentheses may be ignored. Usually $\frac{h_0}{a} < 1$, but h_1 rarely exceeds two to three times a. Thus we have approximately:

$$\frac{\Delta_K}{\Delta_{Kz}} \cong K \sin^2\theta; \tag{206}$$

for example, at $\theta = 45°$,

$$\frac{\Delta_K}{\Delta_{Kz}} \cong \frac{K}{2} \cong 2{,}5-10,$$

i.e. the displacement during rolling of the ship is reduced considerably.

Mounting a machine on inclined mounts not only reduces transverse displacement of a machine during the ship's roll, but also increases the reliability of its attachment and its stability.

This, however, does not exhaust the possibilities of inclined mountings. Analysis reveals that the angle of incline of the supports may have two values at which separation of the vibrations occurs. The use of a large angle, not exceeding 45° however, enables the frequencies

of free transverse and rotational vibrations of the machine to be made nearly the same, and near to the frequency of free vertical vibrations. In principle, these three frequencies may be blended into one, or in any case, they may be contained in a very narrow range, which reduces the possibility of intense resonant vibrations of the machine as mounted, due to the action of forces arising in it, or to the forces of the underway vibrations of the ship.

Because the methods of computation of the frequency of free vibration of machines on inclined supports, calculation nomograms and examples of computations are contained in other works (40, 49), we cannot repeat them in the present work. We may mention merely that mathematical expressions may be obtained not only for inclined supports in two rows, but also for square and elliptical placement (and a mathematical approximation for 8-angle machinery bed-frames). In the latter cases the supports are inclined in two planes.

Despite the undisputable advantage of the method of mounting shipboard machinery on inclined supports which was proposed in 1947, it still has not found extensive application in Soviet shipbuilding practice. This type of vibration isolation of main engines has been incorporated during recent years only on certain ships, where due to the distance between the machine and the sides of the ship additional lateral elastic supports could not be used.

The first publications on this method of vibration isolation in the foreign press (West Germany, 1955) indicate that the method of using inclined mounts for the main and auxiliary machinery of ships is finding very extensive application. In this, the main attention is being focused on reducing the lateral deflection of machinery on inclined supports, but the above-mentioned possibility of using inclined mounts for bringing the frequencies into a narrow range, and "designing" the required frequency spectrum of the machine as mounted is not being taken into consideration.

In foreign installations great loads are permitted on the mounts, but because the vertical stiffness of the mounts with inclined installation of two-layer and similar mounts decreases in comparison with the usual installation, small values of the natural frequency of the mounting, on the order of 5 or 6 cycles, may be obtained very easily. This results in very good vibration isolation, beginning even at 20 cycles. It is difficult to obtain low frequencies with a vertical installation of two-layer vibration mounts because the stiffness of relatively thin rubber inserts working in compression increases with an increase in load on them, and the use of thick inserts provides an elastic attachment which is unstable in a transverse direction.

The design arrangements for attaching inclined mounts to the machine and to the foundation are fairly simple; struts with inclined pedestals are used for this, welded to the machine's bed plate and to the foundation.

52. Sound Isolation of Shafting and Piping Attached to Machines Mounted on Vibration Isolation Mounts

Considerable sound energy may be transmitted by the non-supporting connections of machines, such as drive shafts, air and fluid ducts and pipes, and rigid cables. This energy often exceeds the energy transmitted to the foundation through the vibration isolation mounts. Because of this, non-supporting connections also must be isolated against sound (Figure 156). Flexible sound isolating joints (1a) are installed on drive shafts. The conditions for good vibration

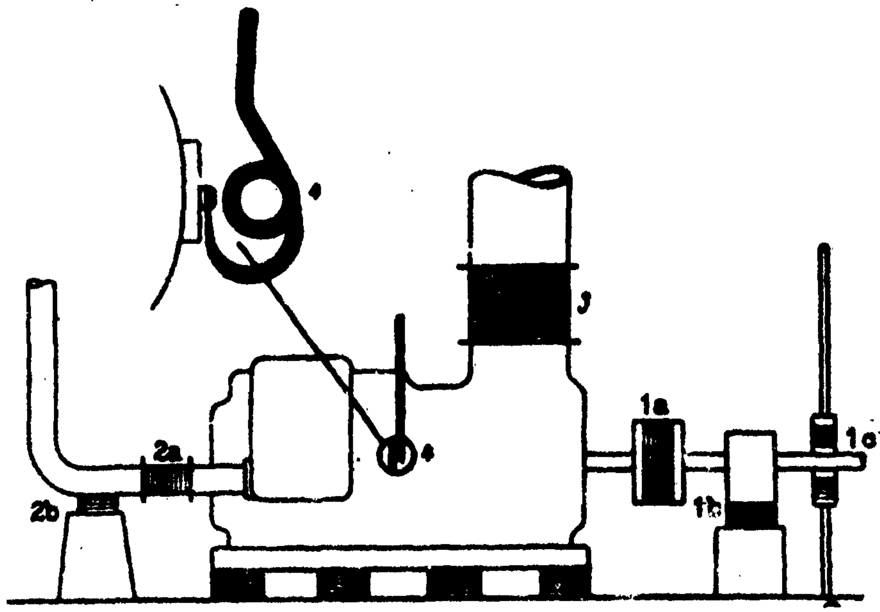
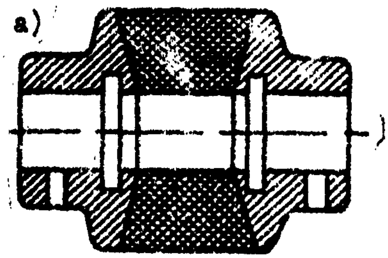


Figure 156. The isolation of drive shafts, pipes and cables of a machine mounted on vibration isolation mounts.

isolation also require the installation of an isolating element under the shaft bearings (1b), especially in cases in which the bearing is located between a machine mounted on vibration isolation mounts and an isolating shaft connector, or when it is the source of sonic vibration (roller bearing). At the point where a drive shaft passes through a bulkhead of the living spaces, a sound-isolating stuffing box-bearing is installed in the bulkhead (1c).

Sound isolating devices for pipes include flexible sound-insulating sleeves for water and oil pipes (2a), sleeves for ventilation ducts (3) and sound isolating supports under pipes (2b). At locations where cables attach to the machine, they are curved in a spiral or loop (4) to reduce the transmission of vibrations by the cable and to prevent damage to the cable due to vibration.

a)



b)

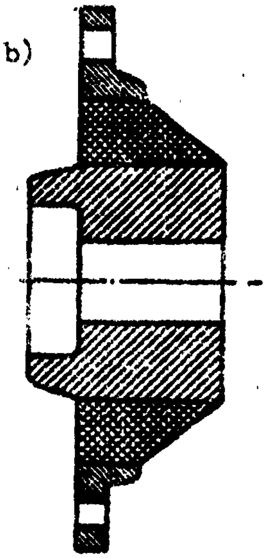


Figure 157. Vulcanized rubber-metal elements of sound isolating couplings, models RSMF (a) and RSMK (b).

Vulcanized rubber-metal connectors are used for isolation of the piping of relatively low power marine, and other engines mounted on isolation mounts, in which the rubber element is vulcanized to the metal parts of the mount. The rubber element is located either along the axis of the coupling (Figure 157-a) or along its radius (Figure 157-b). The normal series of the first type of coupling (type RSMF) consists of eight type-sizes, intended for transmission of torsional moments of 10 or 20, to 15,000 kG/cm. The second series (RSMK) consists of four type-sizes, for torsional moments of 500 to 15,000 kG/cm.

Rubber-metal couplings are able to withstand great loads without injury (Figure 158). In some designs of couplings, rubber sections working in compression are introduced ("Metalastik" coupling, Figure 159) to avoid tearing away of the elastic element from the metal flanges.

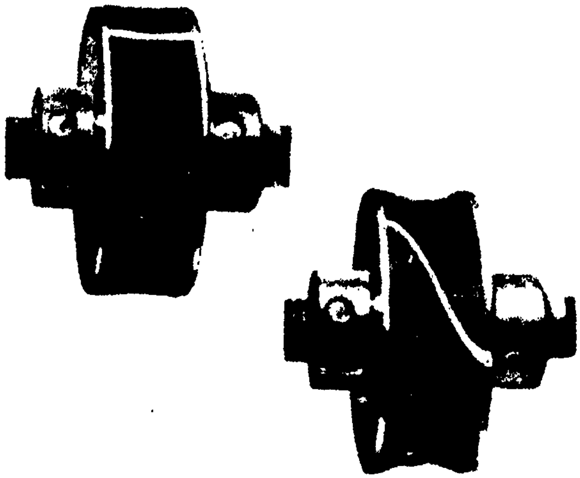


Figure 158. Deflection of the elastic element of sound isolating coupling under heavy load.

Figure 159. "Metalastik" rubber-metal coupling.

Built-up assemblies of rubber inserts, or rubber-metal elements are used in the elastic couplings of the drive shafts of main engines. Pneumatic tire couplings are used extensively in the USSR and abroad, in which the isolating element consists of a detachable inflated rubber ring of rubber-impregnated cloth. When the ring is inflated with air it expands and forms an adequate coupling with the flanges of the drive shaft due to friction. This coupling permits a relative displacement of the ends of the joined shafts to reach 1.5 or 2 mm.

Isolation of air ducts and low-pressure fluid pipes is accomplished usually with the aid of durite sleeves and hoses, secured with the aid of the appropriate clamps and rings. Special-design rubber-metal sleeves may be used for the isolation of high-pressure piping. One of these types of sleeves, type RMSP-T, is shown in Figure 160. The sleeve is designed for a water pressure up to 32 atm, and an oil pressure up to 16 atm.

Hot pipes, such as the exhaust pipes of diesel engines, cannot be insulated with the aid of any type of rubber sleeves. Here sil'fons, elastic corrugated metal sleeves, are used. Due to their

great axial and radial flexibility they isolate low frequency vibrations fairly well. At high frequencies, wave transmission of sound through the metal walls of these sil'fons is possible. Such transmission is reduced through the application of heat resistant, vibration-absorbing layers of material such as various types of asbestos coating (Figure 161-a) to the sleeve. The use of thin-walled sil'fons for insulation of water and oil pipes is not excluded. In this case heat resistance is not required of the vibration-absorbing coating, and it may consist of rubber or plastic composition.

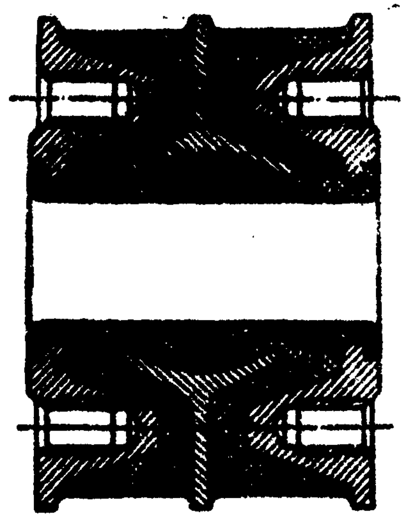


Figure 160. Vulcanized rubber-metal sleeves for high-pressure water and oil pipes.

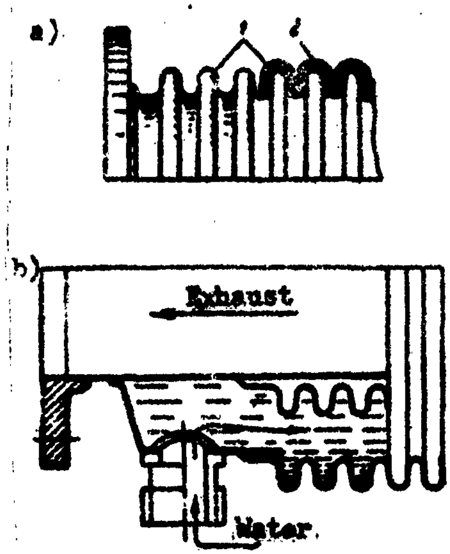


Figure 161. Metallic vibration-isolating couplings (sil'fons) for pipes connected to machinery mounted on vibration isolation mounts.
1 - Body of sil'fon; 2 - Vibration-absorbant coating of sil'fon.

Occasionally sil'fons with water jackets (Figure 161-b) are used for isolation of exhaust pipes. From the point of view of acoustics these sil'fons are better than single-wall sil'fons which do not have a vibration-absorbing coating.

53. The Measurement of the Acoustic Effect of Shipboard Vibration Isolation Treatments

The vibration-isolation effect of anti-vibration mounts is best determined by the experiment diagrammed in Figure 162. In this experiment the airborne noise level is measured in a compartment, one of the boundaries of which is sequentially subjected to excitation by a sonic vibration source not mounted on vibration isolation mounts (left), and one mounted on such mounts (right). The acoustic effect of the isolation treatment may be evaluated on the basis of the difference in noise level in the two cases.

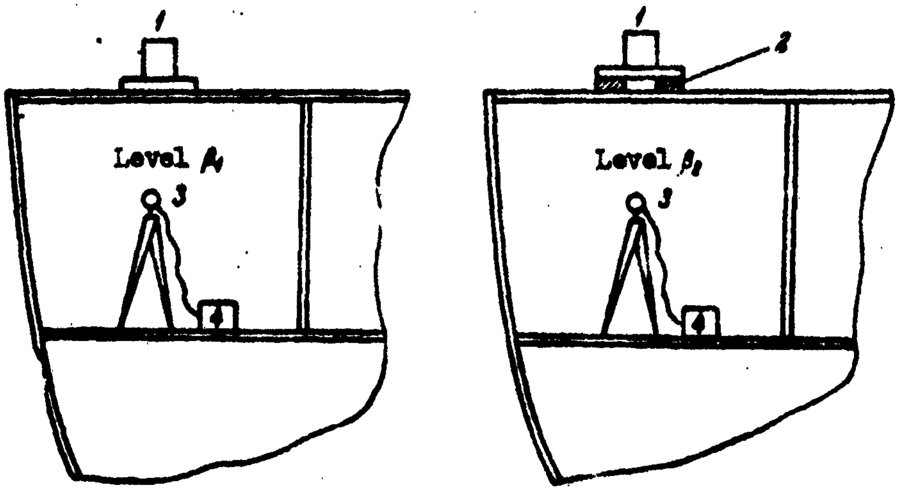


Figure 162. Diagram of an experiment for determining the effect of the sound-isolation mounting of a machine mounted on a deck, upon the noise level in a compartment situated below the deck.
1 - Machine; 2 - Mounts; 3 - Microphone; 4 - Noise meter.

For example, from the results of an experiment of this type (Figure 163) it may be seen that of the designs of built-up vibration isolation mounts tested in the given case, the best vibration isolation was provided by mounts made of several layers of soft, perforated rubber.

However, the direct experiment described cannot be performed in all cases. Most often the acoustic effect of mounts already in place under machines must be evaluated. The method of approximate evaluation of the vibration isolation of mounts through measurement of the drop in sound vibration level in them may be of assistance in this case

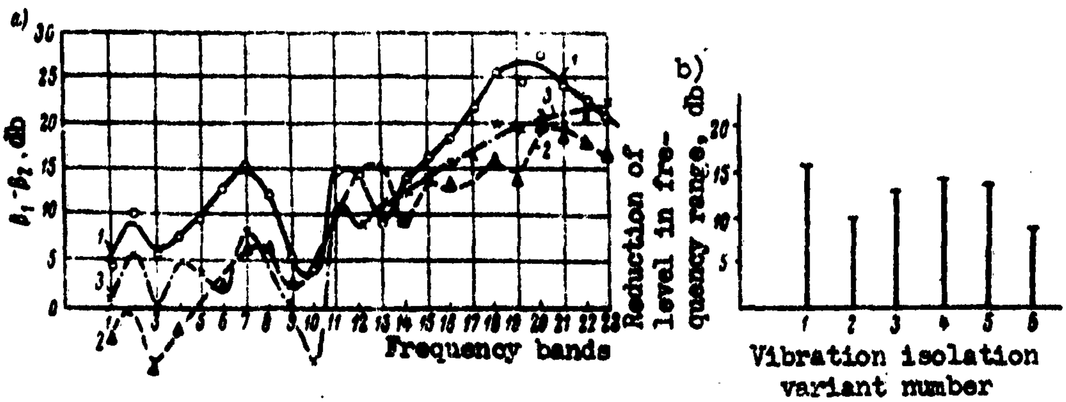


Figure 163. Reduction in the noise level in a below-deck compartment with various types of vibration isolation mounting of a machine on the deck (Figure 162): a - Frequency characteristics; b - Average value of the reduction in the sonic frequency range.
Vibration isolation variants: 1 - Four mounts, 6 x 6 x x 3.2 cm, consisting of 8 layers of perforated rubber; 2 - Four mounts, 6 x 6 x 1.2 cm, consisting of 4 layers of monolithic rubber; 3 - Four mounts, 6 x 6 x 1 cm, consisting of a single layer of felt; 4 - Four mounts, 6 x 6 x x .2 cm, consisting of 8 layers of perforated rubber; 5 - Continuous elastic foundation 3.2 cm thick, consisting of 8 layers of perforated rubber; 6 - Same, 2 cm thick, consisting of 5 layers of monolithic rubber. Weight of the machine 50 kG, size of base 40 x 60 cm.

(Paragraph 47). From the point of view of the technique of measurement, the determination of the drop in vibration levels differs in no way from the measurement of the isolation of light partitions in relation to air noise, i.e. either the method of subsequent measurement of vibration levels at the leg (or bed-frame) of the machine and at the foundation, or the method of simultaneous measurement of the levels of paired circuits connected to a pen-recorder, to a logometer or to a similar instrument may be applied (cf. Figures 105 and 106).

The measurement of vibration levels is conducted at fixed frequencies or in certain frequency bands, and because of this the measured data relative to velocity of vibration and acceleration of vibration practically coincide. In practice, preference must be accorded to measurement of vibrational acceleration levels (at least when working with piezoelectric vibrometers), because these levels greatly

exceed the levels of electrical interference (noise) in the measuring circuits, more so than the levels of velocity of vibration.

As a rule, shipboard vibration isolation may be considered satisfactory if the average drop in vibration levels in the sonic frequency range exceeds 15 to 20 db at the mounts. Here, the actual vibration isolation of the mounts, determining the reduction of airborne noise in the neighboring rooms, varies as a function of different factors in a range of from 5 or 8 to 12 or 15 db.

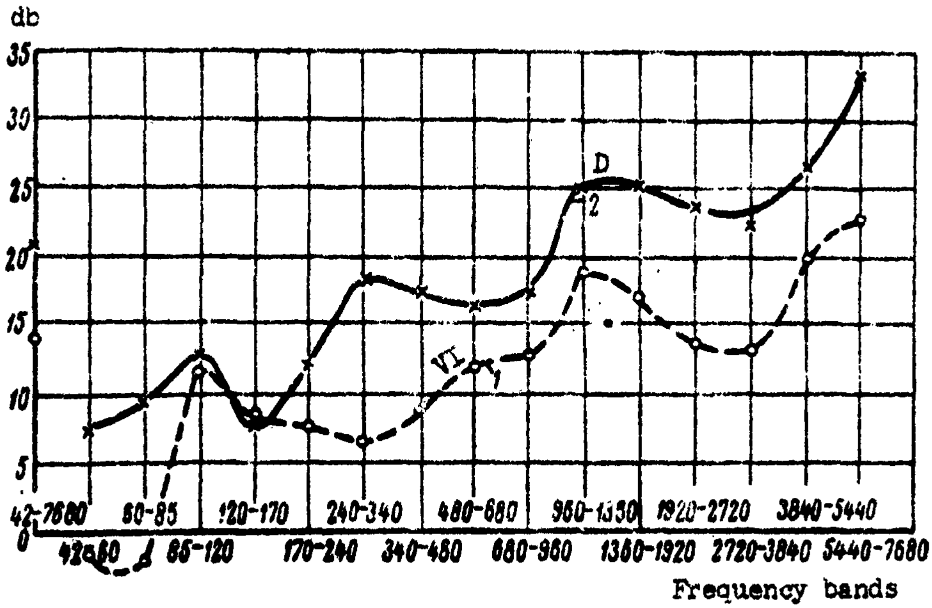


Figure 164. Frequency function of the vibration isolation of vibration isolation mounts (curve 1) and the reduction or drop in sound levels at them (curve 2).

Figure 164, obtained from acoustic investigation of an electric transformer mounted on vibration isolation mounts in the engine room of a ship, gives a picture of the relationship between the reduction in sound level and the vibration isolation at various frequencies. It may be seen from the figure that the difference between the noise reduction and the vibration isolation at medium sonic frequencies reaches 8 to 10 db.

As follows from theoretical considerations (Figure 141), vibration isolation acquires negative values at low sound frequencies, near to the natural frequency of the machine as mounted. As mentioned earlier, the mass of the foundation on which the vibration isolated

machine is installed is of prime importance to the degree of difference in the noise reduction and vibration isolation.

Example 53. On the basis of the conditions of Example 48, determine the amount by which the drop in sonic vibration at the vibration isolation mounts exceeds the vibration isolation of the mounts when installed on a massive, and on a light foundation.

Solution. On the basis of Figure 143 we find that for the massive foundation $\left(\frac{m_\Phi}{m} = 5\right)$: D - VI ≅ 16 db.

The same for the light foundation is $\left(\frac{m_\Phi}{m} = 0{,}5\right)$: D - VI ≅ 4 db.

In the latter case, as was seen from Example 48, the vibration isolation of the mounts is low. In this case, however, the drop in sonic vibration at the mounts more nearly characterizes their true vibration isolation.

The vibration isolation characteristics of flexible couplings and sleeves are evaluated on shipboard in a similar manner, according to the amount of the drop in sound levels. The amount of the drop in sonic vibration at the isolating elements of couplings and sleeves is less than at vibration isolation mounts. In the first approximation it may be considered that sound isolating couplings and sleeves are satisfactory if the drop in the sonic vibration level at them in the axial and radial directions exceeds 10 or 15 db. A more precise evaluation of the acoustic effect of couplings and sleeves requires taking into account the position of the piping relative to the boundaries of living compartments, and the sound isolating characteristics of the vibration isolation mounts.

CHAPTER 14

VIBRATION ABSORPTION

54. Vibration-Absorbing Coatings for Foundations and the Hull Structure of Ships

Another method of controlling sonic vibration on shipboard has become fairly widespread in recent years; it is the use of vibration-absorbing or vibration damping coatings. The acoustic effect of such coatings is based on the introduction of additional damping into the metallic elements of foundations and hull structural components, as a consequence of which the amplitude of the travelling and standing sonic vibrations in them is reduced. According to published data (166, 171), all the hull structure directly adjoining the machinery compartment in some German ships -- walls, decks, bulkheads -- are being covered with vibration-absorbing coatings; this includes also the surfaces on which absorbers for airborne noise are installed (Figure 165).

According to their construction and principle of operation, all the vibration damping coatings may be divided into two groups, hard and soft. Both of these are produced from materials characterized by higher internal friction losses, but the materials of the first group of coatings have a higher modulus of elasticity than the second. Such materials usually consist of hard plastics, frequently with fillers, which further increase the elastic modulus of the coating and also reduce its specific gravity. The coatings of the second group can be produced from soft rubber, plastics, bituminized felt, etc.

The difference in the type of materials used for the coatings determines the difference in the nature of their acoustic effect. When metal plates are covered with hard coatings, their vibrations are damped primarily by the deformation of the coating in a direction parallel to the plate surface. It can be shown that in this case, the vibration absorption by the coating, that is, the reduction of the plate's vibrations within a certain frequency range, should depend very little on the frequency (curve 1 on Figure 166).

In soft vibration-absorbing coatings, the damping is determined primarily by the deformations of the coating along its thickness

dimension. Appreciable damping occurs only at those frequencies at which the length of the elastic wave in the coating is commensurate with its thickness. In view of the low elastic modulus of this type of coating, the wave length becomes comparable with the coating thickness at frequencies of several hundred cycles. The acoustic effect of the coating begins to be felt at those frequencies. In regard to the most unpleasant high frequency oscillations, this effect can, the coating thicknesses being equal, exceed the effect of an acoustic coating of hard vibration-absorbing material (curve 2 on Figure 166). Moreover, taking into account the simplicity of the process of applying soft vibration-absorbing coatings, it should be pointed out that these coatings are just as suitable for use on shipboard as are the hard ones. For example, in the passenger ship "Stuttgart" built in 1960, a soft vibration-absorbing coating was placed over a hard damping layer, which produced an added vibration-absorbing effect of 7-8 decibels at frequencies over 100 cycles per second (170).

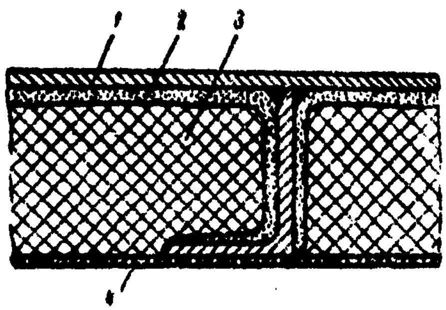


Fig. 165. Covering a ship's hull structure with a sound absorbing coating. 1 - hull plating; 2 - vibration-absorbing coating; 3 - absorber for airborne noise; 4 - perforated screening.

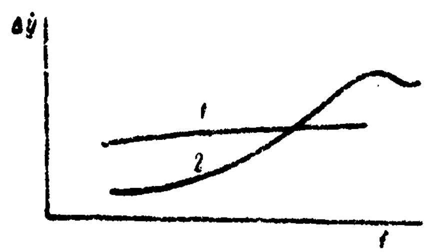


Fig. 166. Characteristic frequency relationship for the reduction of metal plate vibration by the use of hard (1) and soft (2) vibration-absorbing coatings. Vertical axis: acceleration ($\Delta\ddot{y}$).

The effect of any vibration-absorbing coating is observable only at the resonance frequencies of the supporting metal structure. Outside of resonance, there is practically no damping of vibration by the coating. Let us take a closer look at vibration-absorbing acoustic coatings (especially hard ones), the material of which is characterized by greater internal losses. The coefficient of internal losses, which is a measure of the vibration amplitude at resonance of any elastic system, can be expressed as the ratio of the half width Δf of the resonance curve to the resonant frequency of the system f_0:

$$\eta = \frac{\Delta f}{f_0}. \tag{207}$$

By Δf is meant a frequency band equal to half the width of the resonance curve of the system at those points where the vibration amplitude of the system drops to 0.707 amplitude at resonance. Obviously (Figure 6), the greater Δf, the greater the damping in the system. We shall designate:

η_1 -- the loss coefficient of the metal structure without a coating. In the case of ships' foundations and hull structure, the value η_1 varies, at sonic frequencies, within the limits of 10^{-1} to 10^{-2}. A relatively high loss coefficient is characteristic of damping at points of plate joinings, stiffener attachments, and welded seams, etc.;

η_2 -- coefficient of loss in the coating;

η_3 -- coefficient of loss in a metal structure covered by a coating.

The reduction in the amplitude of resonance vibrations of a metallic structure after covering it with a vibration-absorbing coating is

$$\Delta y = 20 \log \left(\frac{\eta_1 + \eta_3}{\eta_1}\right) \text{db}. \tag{208}$$

In the case of metal plates far removed from the source of vibration, the numerical coefficient in formula (208) is less than 20.

It has been shown (156) that in the case of a metal plate covered with a vibration damping coating, the coefficient of loss is equal to:

$$\eta_3 = \eta_2 a \varphi(\xi). \tag{209}$$

Here $a = \frac{E_c}{E_m}$, $\xi = \frac{\delta_c}{\delta_m}$. By E_c and E_m is meant the elastic moduli of the coating and the metal, and by δ_c and δ_m the corresponding values of the thickness of the coating and supporting structure. The nature of the function φ becomes clear from the following.

It may be seen from the last formula that the loss coefficient of the coated plate is proportional to the product of the loss coefficient in the coating multiplied by the elastic modulus of the coating material. The development of materials designed to meet that

requirement is fairly complicated. There are references in the literature to plastic materials with a 0.5 loss coefficient at sonic frequencies and a Young's modulus up to 10^4 kG/cm^2. The specific gravity of such material does not exceed 0.5-0.7.

The relationship of the ratio $\frac{\eta_a}{\eta_a}$ as a function of the variable ξ and the parameter a is shown in Figure 167. When the a and ξ values are fairly high, the $\frac{\eta_a}{\eta_a}$ curves tend toward unity, that is, the loss coefficient of the coated plate approaches that of the coating. For each value of a, there is a value of ξ whereby a further increase in ξ, that is in the coating thickness, has little effect from the point of view of damping the plate vibrations. In ordinary stiff vibration-absorbing materials, the a value varies from 10^{-3} to 10^{-2}. With such a values, as may be seen from Figure 167, an increase in the shield thickness above 3-5 metal thicknesses is impractical. It is considered adequate if the thickness of the vibration-absorbing coating is 2-2.5 times greater than that of the metal upon which it is placed.

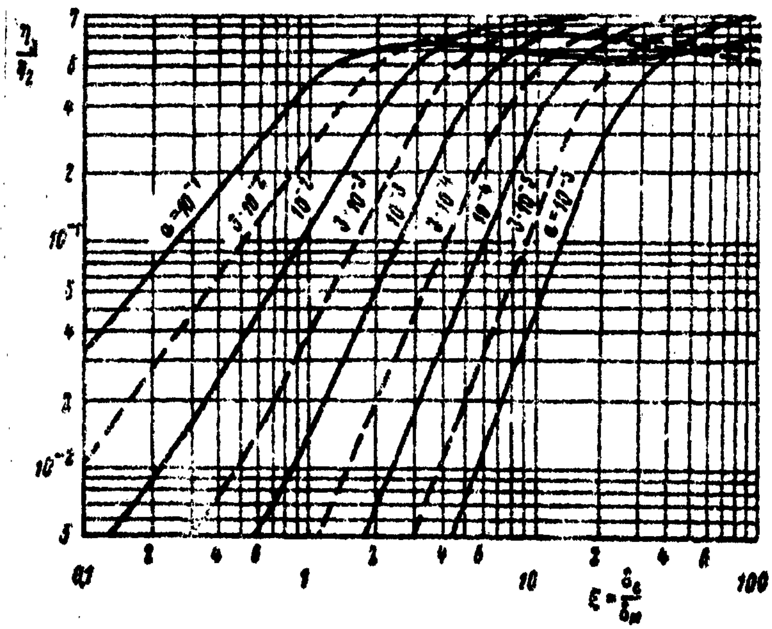


Figure 167. The relative loss coefficient of a coated metal plate as a function of the ratio of the thicknesses of the coating and metal and of the ratio of the moduli of elasticity of the coating and metal.

Corresponding to the above-mentioned values of a and ξ are the approximately linear segments of the $\frac{\eta_3}{\eta_1}$ curves, and the function $\varphi(\xi)$ may consequently be represented as follows (bearing in mind that the curves are drawn in a double logarithmic scale):

$$\varphi(\xi) \approx \xi^2. \qquad (210)$$

Thus in the case of materials and thicknesses acceptable for a coating, the loss coefficient of a metal plate covered by a hard coating is:

$$\eta_3 \cong \eta_2 \frac{E_c}{E_M}\left(\frac{\delta_c}{\delta_M}\right)^2. \qquad (211)$$

Example 54. The loss coefficient of a hard vibration-damping coating at a certain frequency is $\eta_2 = 0.3$, and the elastic modulus of the material $E_c = 2 \times 10^4$ kG/cm^2. The coating is placed over a metal plate whose loss coefficient does not exceed $\eta_1 = 10^{-2}$. The coating thickness should not exceed twice the thickness of the plate. Determine the decrease in the amplitude of resonance vibrations of the coated plate.

Solution. We find the ratio of the elastic moduli of the coating and the plate:

$$\frac{E_c}{E_m} = \frac{2 \cdot 10^4}{2 \cdot 10^6} \cong 10^{-2}.$$

Bearing in mind that $\frac{\delta_c}{\delta_M} = 2$, we shall determine by formula (211) the loss coefficient of the plate when covered by a coating:

$$\eta_3 = 0,3 \cdot 10^{-2}(2)^2 = 0,012.$$

According to formula (208), the decrease in the vibration amplitude of the plate is

$$\Delta y = 20 \log \frac{0.01 + 0.012}{0.01} = 7 \text{ db}.$$

It is possible to combine hard and soft coatings of elasto-plastic materials as well as to make coatings with hard reinforced

casings that increase the elastic modulus and thereby also the vibration-absorbing effect of the coating.

Heat-insulating devices on shipboard made from cork, "ekspanzit", asbestos wood fiber, etc., have definite vibration-absorbing properties. The acoustic effect of these materials is less than that of the special vibration-damping coatings. However, when the heat insulation is 5-10 times as thick as the metal wall on which it is placed, it is possible to obtain a reduction in the sonic vibration of the wall by 6-8 decibels (see Table 24 below) at frequencies over 1 kilocycle per second.

Table 24

The Damping of Bending Waves in a Metal Plate Coated on Both Sides by Heat-Insulating or Vibration-Absorbing Materials

(Plate thickness 2 mm)

Material	Thickness of material on each side of plate, mm	Damping, decibels per meter, in frequency bands, cps.		
		600-1200	1200-2400	2400-4800
Asbestos-wood fiber..	25	10	12	10
Rock wool............	70	5	6	7
Brown cork...........	28	12	16	28
Polystyrene..........	25	14	16	29
Foam layer (perchloro-vinyl.	16	16	18	20
Heat-insulating card-board (dense).......	9	15	16	26
Semi-hard rubber.....	4	12	18	30

Depending on their material composition, vibration-damping coatings can be attached to foundations and hull structure as well as to individual elements of shipboard machinery and systems by gluing, puttying or spraying them on. From the point of view of ship technology, the latter two methods are preferable as they make it possible to coat complicated surfaces. But these methods require the use of special nonshrinking (during their hardening) materials otherwise a thick coating may peel off the metal.

The use of vibration-damping coatings on shipboard for the control of vibration in foundations and hull structure is to some extent

a palliative measure. A better solution would be to produce such structures from materials which poorly conduct sonic vibrations, that is, from plastic materials having high damping and which would be strong enough for structural purposes. There is no doubt that the present level of development in the chemical industry in this field has opened up considerable possibilities. One example is the method used by the Soviet industry in producing plastic hulls for river vessels of medium displacement.

55. Experimental Research on Vibration-Absorbing Materials and Coatings

Elastic and dissipative permanent materials are being investigated with a view to selecting the most suitable materials for vibration-absorbing coatings. One of the most common methods of such investigation is based on determining the degree of the change in the parameters of the resonance curves of thin metal plates and bars when covered with a vibration-absorbing material (27, 156).

An "infinite" bar or plate, coated in sections by the material under test, can be used to determine the magnitude of the damping of flexural waves in metal structures per unit of length. Shown in Table 24 are the values obtained by the author, through the use of this method, of the damping per meter of travelling flexural waves in the plate by covering it with some heat-insulating and sound-absorbing materials as well as materials that could be used for vibration-absorbing purposes.

The thickness of the metal plates in this case is small, only 2 mm, and the damping values per meter are therefore high. This makes possible a more accurate relative evaluation of the vibration-absorbing characteristics of the materials. It is obvious that even a thick layer of sound-absorbing material of the rock wool or asbestos wood fiber type does not increase the damping of sonic vibration. Cork and polystyrene are a little better. A greater attenuation of sonic vibration in metal plates can be achieved by the use of a thinner layer of semihard rubber or plastic at frequencies over 1,000 cps.

Soviet investigations have shown (78, 100) that bituminized felt covered with a mastic on an asbestos base also makes a good vibration-absorption material. Its damping effect can be observed at frequencies below 1,000 cps.

The above discussion of vibration-damping materials deals exclusively with bending vibrations. Longitudinal waves in metal plates are very poorly absorbed by all the types of acoustic coatings. But longitudinal waves can be a source of flexural waves in the structure lying beyond the plates. This circumstance emphasizes the difficulty

of effective vibration absorption in the ships' foundations having a large number of cross connections, as well as the value of combining vibration absorption with vibration isolation treatments, which considerably weakens such longitudinal waves (Paragraph 45).

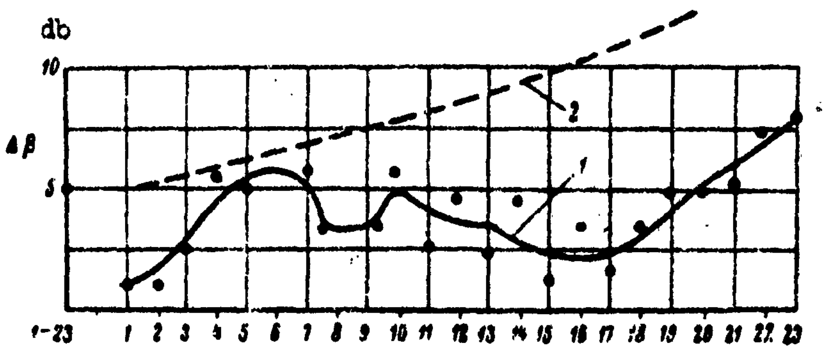


Figure 168. Reducing the noise level in a below-deck compartment by covering the deck with a vibration-absorbing coating in the area of the vibration source.
1 - Source of vibration set on the deck; 2 - Source of vibration set upon a coating (expected curve).

It is considerably easier to weaken the vibrations in the decks, bulkheads and shell plating than in a foundation structure by the use of coatings. Figure 168 shows the frequency characteristics of the noise reduction in a below-deck compartment when a vibration-absorbing coating is placed on a deck around a source of vibration. We should point out the initial damping of a deck with a layer of heat insulation consisting of a 10 mm thickness of granulated cork on a mastic base. Despite that additional damping and the relatively small coating area (1 m^2), the vibration is absorbed to some extent. At high sonic frequencies the noise level can be reduced by 6-8 decibels. Such is the effect when the source of vibration (a hammer vibrator or stamping machine, as it is called) is in direct contact with the deck. When installed on a coating, the vibration absorption effect of the coating is also evident, and the noise reduction below deck should be considerably greater.

Let us look at the effect of vibration-damping layers placed upon the separate parts of a machine and its bed-frame. The possible reduction of the vibrations of a machine, taken as an isolated object, was mentioned previously for this case. But that does not exhaust the role of vibration-absorbing elements. They can improve the isolation

of vibration isolation of the mounts placed under the machine. In this connection their role is entirely analogous to that of the fibrous materials used for the absorption of airborne noise in sound-isolated compartments.

By placing such materials on the inside of soundproof housing (Paragraph 34) we can raise its sound isolation to a value close to that of the housing walls in open space. Without sound-absorbing materials the acoustic effect of the housing is very small, as the continuous operation of the source tends to increase the density of the sound energy within the housing because of the numerous reflections from the housing walls.

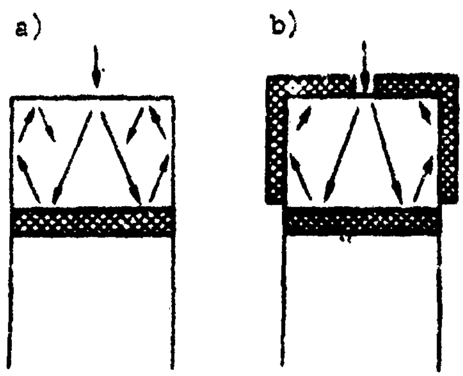


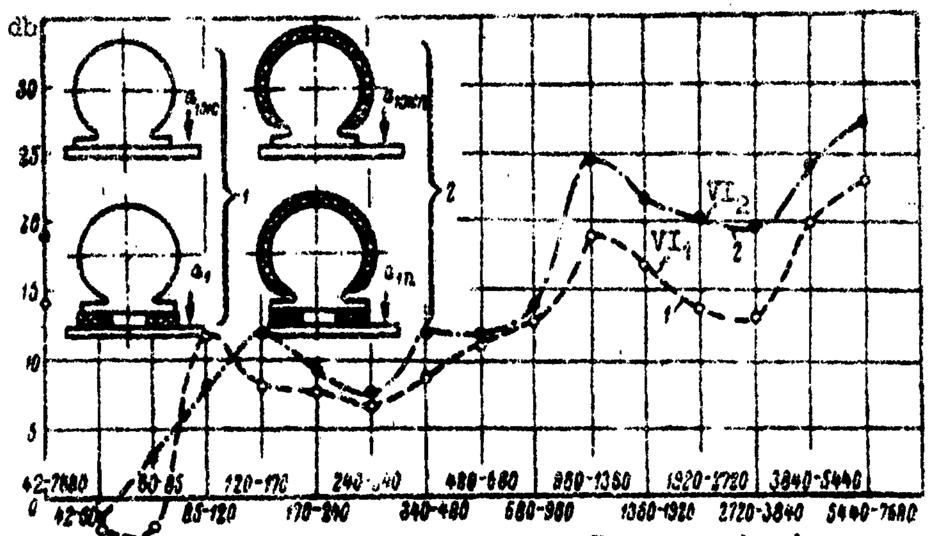
Fig. 169. Determining the effect of coating parts of a machine with vibration-absorbing material, upon the vibration isolation of isolation mounts.

A similar phenomenon occurs in systems where structure-borne sound is isolated by isolation mounts. Although in principle this should apply to any type of vibration isolation mount, it is particularly true of frictionless mounts (steel springs), which merely reflect the vibrations but do not absorb them. In that case, for a small amount of absorption of vibrations in the machine, the density of the vibration energy in it will increase due to the numerous reflections from the boundaries (Figure 169-a). This will cause a corresponding increase in the vibration level also in the foundation structure beyond the mounts, which is equivalent to a reduction in the effective or the true vibration isolation of the mounts in comparison with that value of isolation which could be achieved by installing the mounts on an infinite sound guide [or sound conductor].

However, the density of the vibration energy in the machine can be reduced (Figure 169-b) by covering the boundaries of the machine with sound absorbing materials as a reflector of vibrations. This will raise the vibration isolation effect of the mounts to their full capacity.

An analogy between systems in which airborne and structure-borne sound is propagating is, however, justifiable only to a certain

extent. In vibrational systems, particularly, it is more difficult to separate the absorption of reflected vibrations from the absorption of vibrations in the source itself (machine). This should be borne in mind when evaluating the results of the experiment described below which is designed to determine the effect of a vibration-absorbing coating, placed over various parts of an isolated machine, on the sonic vibration levels of a foundation.



Frequency bands, cps.

Figure 170. The vibration isolation of vibration isolation mounts (1) in the absence of, and (2) with a vibration absorbing coating on the machine.

In this experiment two measurements are taken of the vibration levels at the foundation of a non-isolated and a vibration isolated machine (a shipboard current transformer) for two cases: (1) no vibration absorbing coating and (2) with a coating of plastic on its body. The values of the vibration isolation of the mounts in both cases will be (Figure 170):

$$VI_1 = 20 \log \frac{a_{1ж}}{a_1}; \qquad VI_2 = 20 \log \frac{a_{1ж.п}}{a_{1п}}.$$

Such a coating can be effective at frequencies over 600-800 cycles. The results show that, at these frequencies, for a coated machine, the vibration levels at the foundation of an isolated machine is considerably reduced, and this is equivalent to an increase in the

actual isolation of the mounts. A further increase in the vibration-absorbing effect of isolation mounts can be achieved by introducing a light intermediate foundation structure between the machine and the mounts.

56. Dynamic Vibration Dampers

A vibration-absorbing coating can reduce the flexural vibrations of the hull structure and parts of machines, primarily those of medium and high sonic frequencies. But the vibration of machines as a unit, and also the vibration of framing and plating at the lower natural frequencies, can be very unpleasant. A reduction of that vibration can be achieved by special vibration reducers, or dynamic vibration dampers. They consist of a mass attached to the vibrating system with the aid of an elastic element. The effect of dynamic vibration dampers as a function of their parameters has been investigated by S. P. Timoshenko (102), J. P. Den-Hartog (35) and other authors.

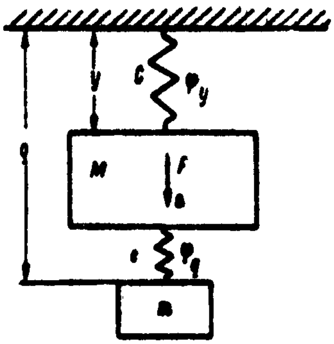


Figure 171. Diagram of a dynamic damper installed in the vibrating system M, C.

Generally, a shipboard machine or hull structure whose vibration is to be reduced is a system with an unlimited number of degrees of freedom. The factor determining the selection of a dynamic vibration damper is that principal vibration of the system whose frequency is close to that of the disturbing force. Therefore by using the method of principal or generalized coordinates (110), it is possible to consider machinery installations and structures as systems with one degree of freedom. Since the simplest type of damper is itself a system with one degree of freedom, the analysis of the action of a dynamic damper is reduced to the solution of the problem of the vibration of a system with two degrees of freedom.

Figure 171 shows a schematic diagram of a dynamic vibration damper m installed on a basic elastic system whose vibrations are to be reduced. The symbols in Figure 171 and in the following text are: c and C represent the stiffness of the elastic element of the vibration damper and of the main system, respectively; M its mass; the harmonic disturbing force applied to the mass has an amplitude F_a. Since the vibration dampers, unlike sound isolation mounts, are characterized by operation at resonance, the friction force in the

damper as well as in the basic system should be taken into consideration. This force is taken into account by the energy absorption coefficients Ψ_q and Ψ_y or by the corresponding coefficients of non-elastic resistance:

$$\gamma_q = \frac{\Psi_q}{2\pi} = \frac{\delta_q}{\pi}; \quad \gamma_y = \frac{\Psi_y}{2\pi} = \frac{\delta_y}{\pi}, \tag{212}$$

where δ_q and δ_y represent the logarithmic decrements of damping.

The coordinates of the mass of the basic system y and of the mass of the vibration damper q are reckoned from the position of their static equilibrium. The differential equations for the motion of the entire system, using the above indicated symbols, can be written as follows:

$$M\frac{d^2y}{dt^2}(1+j\gamma_y)Cy + (1+j\gamma_q)c(y-q) = F_a e^{j\omega t}; \tag{213}$$

$$m\frac{d^2q}{dt^2} + (1+j\gamma_q)c(q-y) = 0.$$

Here, as in the previous chapters, the symbolic form of notation is used; the complex multipliers $(1+j\gamma_{y,q})$ take into account the presence of friction forces in the elastic elements.

Particular solutions of the system (213), describing the steady state vibrations, may be written in the form:

$$y = y_a e^{j\omega t}, \quad q = q_a e^{j\omega t}. \tag{214}$$

Substituting the values y and q and their derivatives in expression (213), we get the following equations after inserting the appropriate transformations for the moduli of the relative vibration amplitudes of the basic system (215), and of the damper (216), as well as the displacement of the mass of the damper in relation to the mass of the main system (217):

$$\frac{y_a}{y_{ст}} = \frac{1}{D}\sqrt{(\chi^2-\varkappa^2)^2+\gamma_q^2\chi^4}; \tag{215}$$

$$\frac{q_a}{q_{ст}} = \frac{1}{D}\chi^2\sqrt{1+\gamma_q^2}. \tag{216}$$

$$\frac{y_a - y}{y_{CT}} = \frac{x^2}{D}; \tag{217}$$

$$D = \sqrt{[x^4 - (1 + \chi^2 + \mu\chi^2)x^2 - (1 - \gamma_y\gamma_q)\chi^2]^2 + [(\gamma_y + \gamma_q)\chi^2 - (\gamma_y + \gamma_q\chi^2 + \mu\gamma_q\chi^2)x^2]^2}. \tag{218}$$

The following designations were introduced in the above expressions:

$y_{CT} = \frac{F_a}{C}$ -- displacement of the main system under the static action of the amplitude value of the disturbing force;

$\mu = \frac{m}{M}$;

$x = \frac{\omega}{\omega_0}$ -- the ratio between the frequency of the disturbing force and that of free vibrations of the main system installed on an infinitely massive foundation (partial frequency of the main system);

$\chi = \frac{\omega_{0q}}{\omega_0}$ -- the ratio between the partial frequency of the dynamic damper the same frequency of the main system. The value χ is occasionally referred to as the tuning of the damper.

Formulas (215) - (218) are basic to the calculation of the frequencies of a dynamic damper and the vibration amplitudes of the main system and damper. Presented in Figure 172 is a chart, based on formula (215), showing the frequency relationship of the vibrations of the main system when vibration dampers with different internal-loss coefficients are installed (internal losses are assumed to be absent in the main system of the damper, that is $\gamma_y = 0$).

The vibration curve of the main system without a damper is not shown in Figure 172. It differs from curve 5 only by the fact that its maximum, to be accurate, corresponds to the resonance frequency whereas in curve 5 it is reached when $\frac{\omega}{\omega_0} < 1$. In the case of a vibration damper without losses that maximum disappears but is replaced by two new resonances (curve 1) with frequencies above and below the resonance frequencies of the main system due to the formation of a system with two degrees of freedom. The frequency "dispersion" of a system with a vibration damper is determined by the relationship between the parameters of the main system and the damper.

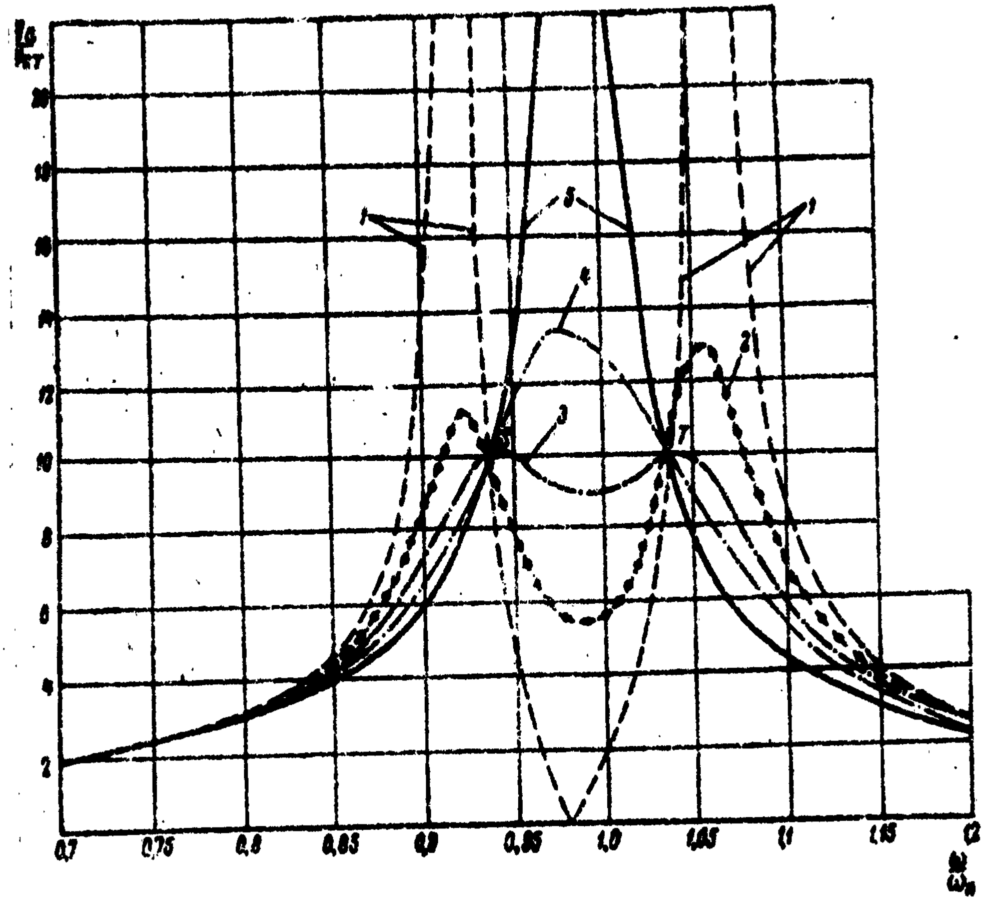


Figure 172. Resonance curves for the main system without friction for various values of the loss coefficient in the vibration damper system.

$1 - \psi_q = 0$; $2 - \psi_q = 0{,}65$; $3 - \psi_{q\,opt} = 1{,}1$; $4 - \psi_q = 1{,}65$; $5 - \psi_q = \infty$.

If even a small amount of friction is introduced into the damper, the vibration amplitudes of the main system at resonances would have finite values (curve 2), as would be expected. For various values of friction in the damper, the vibration amplitude curves intersect at one and the same points, designated as S and T. A value of Ψ_q can be selected in such a way that the $\frac{y_a}{y_{cT}}$ values in the entire range of frequency changes would not exceed the $\frac{y_a}{y_{cT}}$ values at points S and T.

This value of the friction coefficient in a vibration damper is called the optimal; it corresponds to curve 3 in Figure 172.

When the friction is even greater, the two-peak amplitude curve becomes a single-peak curve (curve 4). Finally, with Ψ_g infinitely large, which amounts to a rigid connection between the vibration damper and the main system, the amplitude curve of the main system (curve 5 in Figure 172), as has already been mentioned, becomes similar to the curve of the same system without a damper. In this case the resonant frequency is somewhat lower than ω_0, as the mass of the system has increased from M to M + m.

Let us consider in greater detail the characteristics of dampers with and without friction. To devise a calculation formula for a damper without friction in the (215) - (218) formulas, γ_g is placed equal to zero. Then:

$$\frac{y_a}{y_{cr}} = \frac{\chi^2 - x^2}{D_1}; \tag{219}$$

$$\frac{q_a}{y_{cr}} = \frac{\chi^2}{D_1}; \tag{220}$$

$$\frac{q_a - y_a}{y_{cr}} = \frac{x^2}{D_1}; \tag{221}$$

$$D_1 = \sqrt{[x^4 - (1 + \chi^2 + \mu\chi^2)x^2 + \chi^2]^2 + \gamma_y^2(\chi^2 - x^2)^2}. \tag{222}$$

A detailed analysis of the expression (219) would show that when the damper frequency is tuned to $\chi \approx 1$ and $\mu < 0.05$, the free vibration frequencies of the system with a damper would be located practically symmetrically in relation to the partial frequency of the main system. An increase in μ is conducive to a greater frequency dispersion and a disruption of their symmetric arrangement in relation to ω_0. As χ moves away from 1, the value of one of the new frequencies approaches the partial frequency of the main system, and the other becomes closer to the value of χ.

The maximum vibration amplitudes of the main system with a damper are found in the resonance peak areas and determined by the following expression:

$$\left(\frac{y_a}{y_{cr}}\right)_{max} = \frac{1}{\gamma_y}, \tag{223}$$

that is they are the same as the vibration amplitude of the main system without a damper when $\omega = \omega_0$.

At a frequency equal to the partial frequency of the dynamic damper, that is when $\chi = \varkappa$, there are no vibrations of the main mass. Formally, this applies to any mass of the damper but in the case of very small m values, the vibration amplitude of the damper would be so high that it would destroy the elastic connection. For practical purposes, the mass of the dampers in use is 5%, and in some cases 7-8%, of the main system, that is $\mu < 0{,}05—0{,}08$.

That a dynamic damper without friction can under certain conditions reduce the vibration amplitude of the main system to zero is an indicator of its considerable effectiveness. But such a reduction occurs only in a very narrow frequency band (let us recall the middle branch of curve 1 in Figure 172); it is therefore useful to use a damper without friction in installations with fixed operating conditions or with a very limited range of operating rpm.

The parameters of the dynamic damper, μ and χ, are selected in such a way that the vibration amplitudes of the main mass do not exceed the permissible magnitude in the region of operating revolutions. By designating the permissible relative amplitudes at the lower and upper boundaries of the operating region as a_1 and a_2 (see Figure 173 where curve 2 represents one of the segments of curve 1 in Figure 172), it is possible to obtain from equation (219) the required values of μ and χ:

$$\mu = \frac{\left(\varkappa_2^2-\varkappa_1^2\right)\left[1+a_1\left(1-\varkappa_1^2\right)\right]\left[1-a_2\left(1-\varkappa_2^2\right)\right]}{\varkappa_1^2\varkappa_2^2\left[a_1+a_2+a_1a_2\left(\varkappa_2^2-\varkappa_1^2\right)\right]}; \tag{224}$$

$$\chi = \varkappa_1\varkappa_2\sqrt{\frac{a_1+a_2+a_1a_2\left(\varkappa_2^2-\varkappa_1^2\right)}{a_1\varkappa_1^2+a_2\varkappa_2^2+a_1a_2\left(\varkappa_2^2-\varkappa_1^2\right)}}, \tag{225}$$

where $\varkappa_1$ and $\varkappa_2$ are the $\frac{\omega}{\omega_0}$ values at the boundaries of the operating region (Figure 173).

To calculate the strength of the damper's elastic connection, we must know the vibration amplitude of the damper load in relation to the main system. At the boundaries of the operating region these amplitudes are

$$\left(\frac{q_a - y_a}{y_{cT}}\right)_{1,2} = \left|\frac{\varkappa_{1,2}^2}{\left(\chi^2 - \varkappa_{1,2}^2\right)\left(1 - \varkappa_{1,2}^2\right) - \mu\varkappa_{1,2}^2\chi^2}\right|. \tag{226}$$

Such are the basic calculations required to determine the parameters of a damper without friction.

As may be seen from Figure 172 again, the width of the frequency band within which it is possible to reduce the vibration of the main system is considerably wider for the case of a damper with friction than for one without friction, but the degree of vibration damping may be somewhat less.

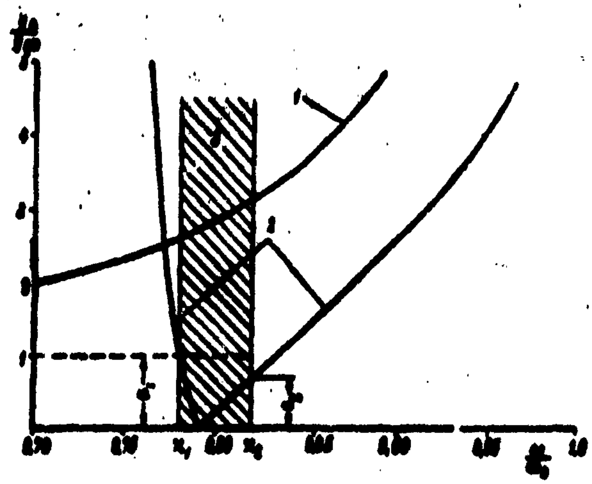


Figure 173. Determining the parameters of a dynamic damper without resistance tuned to the operating region.
1 - without a damper; 2 - with a damper; 3 - operating frequency region.

In the majority of cases it is practical to use a vibration damper with optimal friction. This damper can be adjusted in such a way that the ordinates of curve $\frac{y_a}{y_{cT}}$ at points S and T are the same (curve 3 in Figure 172). The adjustment in this case would be expressed as

$$\chi_{S=T} = \frac{1}{1+\mu}. \tag{227}$$

Without drawing any further conclusions, we shall point out that the value $\frac{y_a}{y_{cr}}$ at points , and T will be equal to

$$\left(\frac{y_a}{y_{cr}}\right)_S = \left(\frac{y_a}{y_{cr}}\right)_T = \sqrt{\frac{2+\mu}{\mu}}, \tag{228}$$

hence

$$\mu = \frac{2}{\left(\frac{y_a}{y_{cr}}\right)_{S,\,T} - 1}. \tag{229}$$

The other values required for calculating a vibration damper with optimal friction are also fairly simply expressed by means of μ.

The optimal value of the coefficient of energy absorption in the vibration damper is

$$\Psi_{q_{opt}} = 2\pi \sqrt{\frac{\mu(3+2\mu)}{2+\mu}}, \tag{230}$$

or, since $\mu \ll 1$,

$$\Psi_{q_{opt}} \cong 7.7\sqrt{\mu}\,. \tag{231}$$

The maximum vibrational displacement of the damper mass in relation to the main mass is

$$\left(\frac{q_a - y_a}{y_{cr}}\right)_{max} = \frac{1+\mu}{\mu}\sqrt{\frac{2(2+\mu)}{3+2\mu}} \cong \frac{1{,}15}{\mu}. \tag{232}$$

The above formulas hold both in the absence of friction in the main system and for any feasible value of friction in it (up to $\Psi_y = 2.5$).

We shall point out some instances of the use of dynamic vibration dampers on shipboard. Thus, in one of the Soviet ships a dynamic damper reduced the vibration of a lubricating oil pump and an in-port generator in their operating region by 3-4 times (see oscillograms in Figure 174). The vibration damper used for that purpose is shown in Figure 175. Horizontal steel rods are used as the elastic elements of the vibration damper for vertical vibrations. At the ends of the rods are fastened masses. They are fastened in such a way as to make it

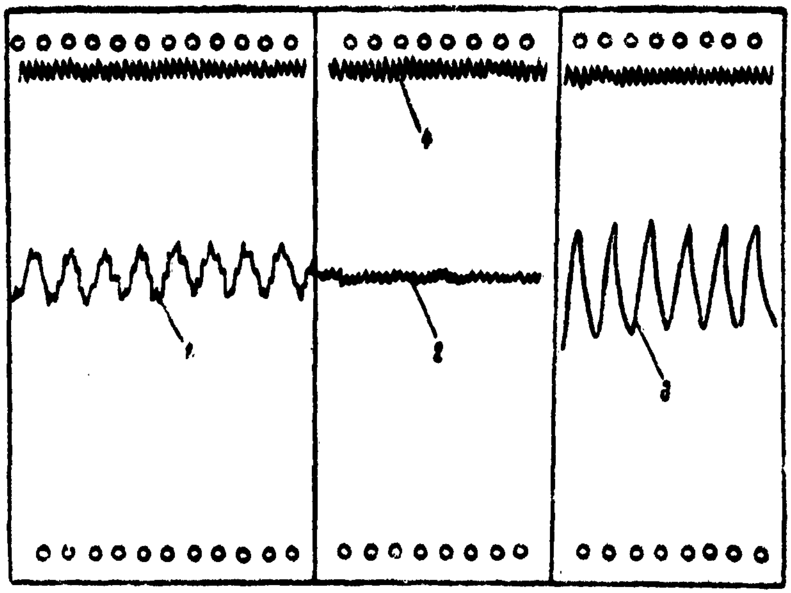


Figure 174. Oscillograms showing the vibrations of a marine lubricating oil pump.
1 - without a vibration damper; 2 - with a vibration damper; 3 - vibration of the damper weight; 4 - time divisions.

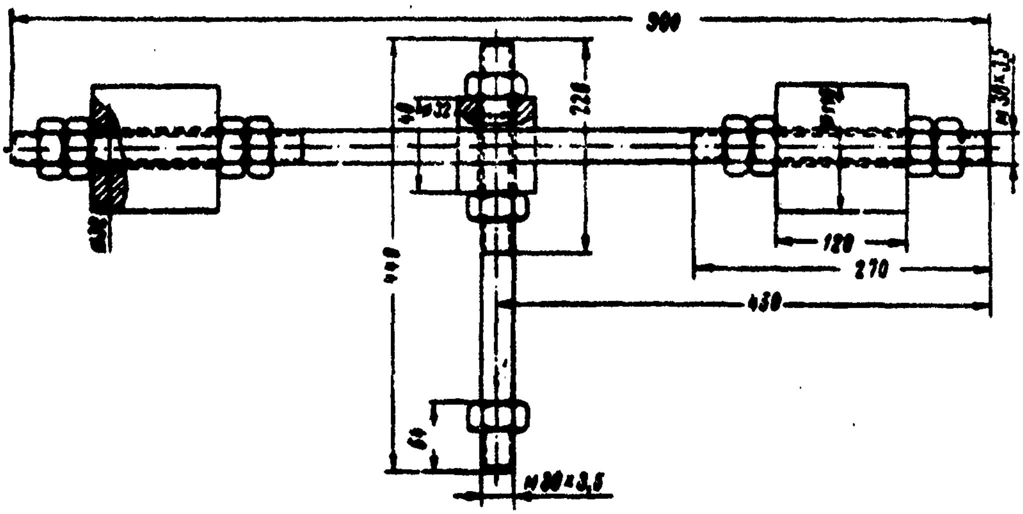


Figure 175. Dynamic damper with two degrees of freedom for a diesel generator.

possible to regulate their arms and to tune the vibration damper on the spot. The design and theory of this vibration damper were developed by A. M. Alekseyev and A. K. Sborovskiy.

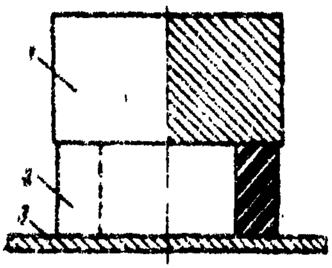


Fig. 176. A rubber-metal damper for the shell plating of a ship. 1 - damper weight; 2 - elastic rubber connection (ring); 3 - shell plate.

The vertical rod of the damper is an elastic element designed to absorb the rotary vibrations in the transverse plane of the main system. Thus this vibration damper has two degrees of freedom and reduces vibration simultaneously in a vertical direction and in the transverse plane, as was borne out by a test.

Another instance of practical utilization of dampers is their use to reduce the intensive resonant vibration of the shell plating in powerful pusher-tugs when their main engines are in operation. The amplitude of that vibration was as high as 1 mm and the frequency was 45 cps. [Note: cps is an abbreviation for cycles per second 1 cps = 1 hertz.]

Vibration dampers with friction, the design of which was proposed by the author, consisted of metal plates glued to circular rubber rings (Figure 176). The rubber rings in turn were glued to the middle of the spaces between frames. The weight of each damper amounted to 3.3 kilograms. A total of 30 dampers were thus installed, one in each frame bay.

As a result, the vibration of all the shell plates without exception, was reduced to 0.1 mm, that is by ten times. This experience clearly illustrated the value of using dampers on shipboard. Vibration dampers can be used to reduce intensive vibration of the walls of sound isolating housings, air ducts, bulkheads and decks. Figure 177 shows a vibration damper designed to reduce the vibration of the deck in a passenger's saloon on a ship of 5,000 tons displacement (158). The saloon is located over the machinery compartment, which has two diesel engines with reduction gears. With the propeller shaft rotating at n = 120 rpm, the vibration of the deck, at the eighth harmonic of the shaft rpm and originally amounting to 0.3 mm, was reduced three-fold by the damper (see curves in lower part of Figure 177).

In addition to the linear dynamic dampers with and without friction, discussed in the above paragraph, the following types can

also be used in ships: dampers with nonlinear elastic elements, pendulum-type antivibrators, and dampers whose frequency is automatically tuned to that of the disturbing force. But the description of all these dampers is not within the purview of this book.

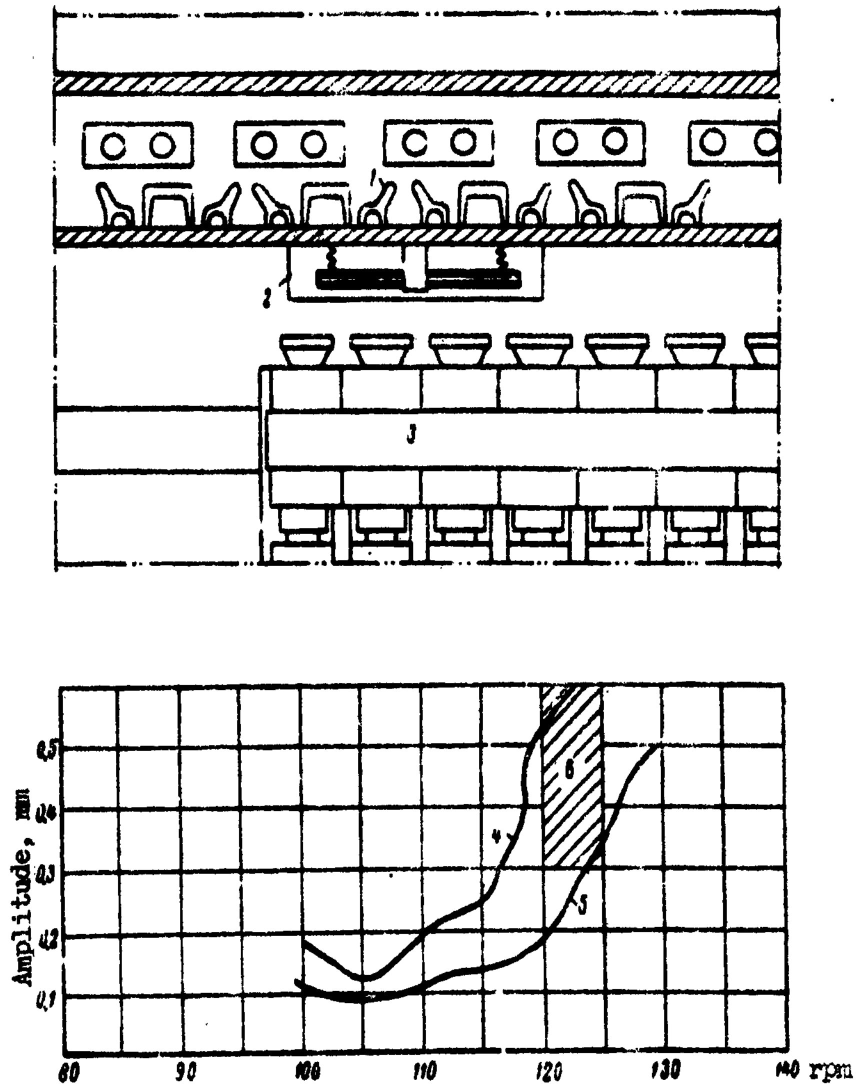


Figure 177. Installation of a dynamic vibration damper to reduce local deck vibration in the lounge area of a passenger ship. 1 - lounge, first class; 2 - vibration damper; 3 - engine compartment; 4 - lounge deck vibration (8th order of propeller shaft revolutions); 5 - same, with the use of vibration damper; 6 - normal operating region.

Example 55. After the installation of a diesel generator on one of the ship's decks there was observed a considerable vertical vibration of the generator. The frequency of the free vertical vibrations of the diesel generator on the deck was $f_0 = 30$ cps, the vibration frequency of the generator in operation was $f = 24$ cps, and the instability in the generator's rpm was $\delta = \pm 2\%$.

Determine the parameters of an antivibrator to be installed so that the vibration of the generator does not exceed $y_a = 0.3$ mm.

Solution. Let us determine the $\varkappa$ values at the boundaries of the operating region of revolutions:

$$\varkappa_1 = \frac{f}{f_0}(1-\delta) = \frac{24}{30}(1-0{,}02) \cong 0{,}78;$$

$$\varkappa_2 = \frac{f}{f_0}(1+\delta) = \frac{24}{30}(1+0{,}02) \cong 0{,}82.$$

By the nomogram in Figure 151 we can determine that the static deflection $y_{ct} = 0.28$ mm corresponds to a free vibration frequency of 30 cps. We assume the relative amplitudes a_1 and a_2 at the boundaries of the operating region (see Figure 173) to be the same. These amplitudes are:

$$a_1 = a_2 = \frac{y_a}{y_{ct}} = \frac{0{,}3}{0{,}28} \cong 1.$$

Since the operating revolutions in this case change within a very narrow range, it would be rational to use a dynamic damper without resistance. Its parameters can be found from formulas (224) and (225):

$$\mu = \frac{(0{,}82^2-0{,}78^2)[1+(1-0{,}78^2)][1-(1-0{,}82^2)]}{0{,}78^2\cdot 0{,}82^2\,[2+0{,}82^2-0{,}72^2]} = 0{,}07;$$

$$\gamma = 0{,}78\cdot 0{,}82\sqrt{\frac{2+0{,}82^2-0{,}78^2}{2\cdot 0{,}82^2}} = 0{,}79.$$

Thus the mass of the damper is 7% of the mass of the generator, and its free vibration frequency is

$$f_{0g} = \gamma f_0 = 0{,}79\cdot 30 = 23{,}7 \text{ cps}.$$

The damper can be designed as indicated in Figure 175.

Example 56. Design a dynamic damper with a rubber elastic element for reducing the vibration of the plating of a bulkhead between stiffeners. The bulkhead has a thickness of 6 mm and a "reduced mass" of 41.5×10^3 grams. [The concept of "reduced mass" is part of the method of principal or generalized coordinates, which is frequently used in the USSR for the analysis of vibration problems. It is explained in reference (110) -- Translator's note.] The free-vibration frequency of the plating in the between-stiffener bays is 45 cps, the vibration amplitude of the supporting contour of each panel is 0.03 mm and the permissible vibration amplitude of the plating is 0.15 mm.

Solution. If we assume that the friction in the rubber element of the damper is optimal, we may utilize formulas (227) - (232). Since $\left(\frac{y_o}{y_{ct}}\right) = \frac{0.15}{0.03} = 5$, the value of μ, according to formula (229), equals

$$\mu = \frac{2}{(5)^2 - 1} \cong 0.08 .$$

Hence the mass of the damper is

$$m = \mu M = 0.08 \times 41.5 \times 10^3 = 3.3 \times 10^3 \text{ grams.}$$

The proper tuning of the damper is determined by formula (227):

$$\chi = \frac{1}{1 + 0{,}08} = 0{,}925.$$

The partial frequency of the free vibrations of the damper is:

$$f_{o_q} = \chi f_o = 0{,}925 \cdot 45 \cong 42 \text{ cps.}$$

We shall select the design of the damper according to Figure 176. These are the initial dimensions of the rubber element: height 40 mm, external diameter 95 mm, internal diameter 65 mm. The static modulus of elasticity of the rubber (soft) is 15 kG/cm^2, and the coefficient of dynamic stiffness at 45 cps. is 1.5. Since the area of the free lateral surfaces of the rubber is large, the stiffness coefficient of the boundary (Figure 152) does not exceed 1, and the stiffness of the rubber element can be calculated according to Hook's law. This stiffness is equal to

$$C \approx \frac{15 \cdot 1{,}5\pi\,(9{,}5^2 + 6{,}5^2)}{4 \cdot 4} \approx 210 \text{ kG/cm}.$$

The free-vibration frequency of the vibration damper is

$$f_{0g} \approx \frac{1}{2\pi}\sqrt{\frac{210 \cdot 10^6}{3{,}\text{[illegible]} \cdot 10^3}} \approx 40 \text{ cps}.$$

In view of the possible errors in determining the elastic modulus and dynamic stiffness, the final adjustment of the stiffness of the elastic element is done experimentally.

The required optimal value of the energy absorption coefficient in the vibration damper (formula 231) is:

$$\Psi_{opt} \approx 7{,}7\sqrt{0{,}08} \approx 2{,}2.$$

Actually it was found to be a little less ($\approx$ 1.6).

Since the mass of the damper is relatively large (8% of the main mass), there are no grounds for expecting excessive vibration amplitudes of the damper, nor for that reason any rupture of its elastic element.

Dynamic vibration dampers are not the only possible methods of reducing the low frequency resonance vibration of metal plates. In principle it is possible to use for such purposes the induction method of vibration absorption based on the inhibiting action of Foucault currents in paramagnetic materials. Such a vibration absorber consists of an aluminum ring attached to a vibrating plate and placed in a strong magnetic field (120). This device made it possible to reduce the resonance vibration of various parts of an airplane fuselage by 15-20 decibels at frequencies of 150-200 cps.

CHAPTER 15

OTHER METHODS FOR REDUCING NOISE AND VIBRATIONS

IN SHIPS

57. Reducing the Vibration Produced by Propulsor Operation and Propeller Shaft Rotation

Modern marine propulsors -- propeller and water-jet types -- can produce an intensive general and local vibration of the ship and noise inside it. In the case of ships equipped with hydrofoils and ice-breakers, the noise can be created by the impact of waves and ice against the bottom and sides of the ship, and in the case of high speed planing craft, noises can be created by air propellers when these are used as propulsors.

Sonic vibrations occasioned by the operation of propellers can be produced by the suction forces of the propeller, cavitation on the propeller blades, and propeller "singing."

The amplitudes of the hydrodynamic suction forces developed on the ship's hull by the propeller's operation can be quite considerable. The methods of calculating the suction forces were developed by N. N. Babayev (16), F. Lewis (147), etc. The main frequencies of the suction forces f_3 correspond to the first order of the number of propeller revolutions and to an order equal to the number of propeller blades, that is formula (233)

$$f_s = i\frac{n}{60}; \quad i = 1;\ m, \tag{233}$$

where n -- is the propeller shaft rpm;

m -- the number of propeller blades.

When these frequencies coincide with one of the free-vibration frequencies of the ship's hull (17, 61, 65), they can set up a strong vibration. In view of the low frequencies of propeller vibration, it cannot by itself produce a considerable auditory effect. But the subjective sensation of vibration can be quite unpleasant; moreover, a

low-frequency vibration can produce all sorts of squeaks, rattling of poorly fastened metal structures, and a modulation of the machinery noises.

The cavitation noise of a screw propeller is produced by the closing of air and water vapor cavities appearing on the propeller blades at a certain critical speed. These vibrations are imparted through the water to the shell of the ship, producing a noise in the after compartments that can be subjectively described as "cracking," roaring, "champing," and whistling.

"Singing" of the propeller is a relatively rare phenomenon. It occurs when the frequencies of the disturbing hydrodynamic forces coincide with the natural frequencies of the propeller blades. The singing noise can occasionally be heard through the whole ship. The singing noise can be eliminated by sharpening the leading edge of the blades (127).

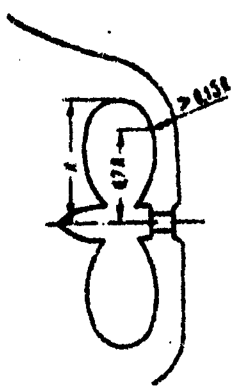


Fig. 178. Determining the permissible clearances between the propeller blades and hull of the ship.

The reduction of the suction forces on the hull as well as the reduction, to a certain extent, of the cavitation effects can be achieved by increasing the clearance between the propeller blades and the hull, particularly with respect to those parts of the propeller furtherest removed from the boss. At a distance equal to 0.7 of the propeller disk radius, that is where the strongest hydrodynamic forces develop, the distance between the edges of the propeller blades and the ship should be at least 0.15 of the propeller radius (Figure 178). It is known from published data also that a sizeable reduction in the hydrodynamic forces acting upon the plating can be achieved by a careful balancing of the propellers and by the use of 5-bladed propellers (116, 147).

Quite interesting is the possibility of reducing the vibrations of the plating in the propeller area by the use of elastic layers. This concept was first voiced by N. N. Babayev in 1945. A vibration control device of that type was installed on the hydrographic ship "Nord" built in Western Germany in 1955. It consists of a 30 mm thick rubber diaphragm fastened in an aperture and fitted flush with the shell plating. It was pointed out (167) that at low and average speeds this device reduced the noise in the after compartments by 4-5 decibels.

Figure 179 shows the device used on Soviet ships. Inside the vibration-prone plating is a trunk filled with water. The hydrodynamic forces produce a vibration of the water surface in the trunk but this vibration is not transmitted to the hull. The trunk may be closed off above the water surface or extended up to the upper deck (dotted line in Figure 179).

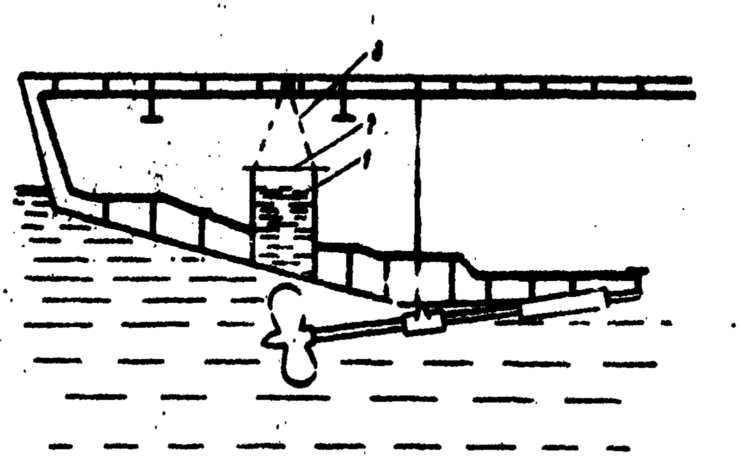


Figure 179. Device designed to reduce the hull vibration caused by the action of the propeller.

1 - water-filled trunk; 2 - trunk cover; 3 - metal enclosure.

From an acoustical point of view, the latter variant is preferable as it prevents the resonant vibration of the air column above the water surface in the trunk. The arrangement shown in Figure 179 made it possible to reduce the vibration of the after part of the ship several times at frequencies of the first and third orders (94). Attempts to achieve a similar reduction in the vibration by other methods had not been successful.

Vibration energy can be transmitted to the plating not only through the water but also through the propeller struts and rudder. The vibration isolation of the rudder stock from the surrounding structure with sea-water resisting rubber inserts facilitates a reduction in the vibration energy transmitted to the rudder blade and thence to the hull.

If none of the above-listed measures is possible or in any way effective against propeller-produced vibrations, recourse should be had

to the means of isolation and vibration absorption inside the ship. To be recommended first of all is an arrangement of sound isolating walls about 50-60 mm from the plating fastened to the frames using elastic gaskets; also, vibration-absorbing coatings on the plating in the after part of the ship. It was suggested that part of the bottom of the river passenger craft "Moskvich" be covered with bitumen (48). The bottom of the diesel ship "Stuttgart" (170) was covered with a 14 centimeter thick layer of bitumen in the area of its cycloidal propeller.

To isolate the propeller noise penetrating into the midship part of the ship through the propeller shaft tunnel, light double sound isolating partitions are installed in the funnel (131). These partitions are removable or fixed, depending on prevailing conditions.

The plating of shallow-draft river boats equipped with water-jet propulsion is subject to considerable vibration and humming especially in turns and when backing, because the streams of water from the jet pipes are directed along the sides toward the bow. In this case, the airborne noise inside the ship can be reduced (as in the case of the noise produced by the impact of waves against the plating of ships equipped with hydrofoils) by fastening soundproof partitions to the plating and bulkheads using elastic materials and leaving a fairly large gap between the partitions and the plating. Similar to this are the sound-isolating devices used on the sides of icebreakers to eliminate the noise inside the ship when travelling through ice.

Improperly balanced shafting on the ship can be a source of intensive first order vibrations. Higher frequencies produce resonant transverse and torsional vibrations of the shafting arising from the presence of a number of elastic and inertial elements in the shaft system. The methods of fighting torsional vibrations are well known; the Soviet school headed by V. P. Terskikh holds a leading place in the field of fighting torsional vibrations in ships' propeller shafting.

Vibration and unpleasant noise of a changing nature, due to the noncoincidence of the shaft revolutions, are occasionally observed on two- and three-shaft ships. This vibration can be eliminated by synchronizing and phasing the shaft rotation.

In air cushion craft, very intense noise can be produced by the air blower which forms the cushion and by the propeller installation that produces the thrust. Effective soundproofing measures must be used to make such craft properly habitable. Such measures include the use of sound isolating housings and vibration isolation shock mounts, ventilation-type silencers, and sound-screening partitions in the vicinity of the air propeller.

58. Reducing the Noise of Water, Plumbing and Other Shipboard Systems

The problem of silencing the noise in ventilation systems was dealt with in Chapter 10. Intense and unpleasant noises may arise also in the water pipelines, the plumbing system, technical water systems, elevators and hoists.

There are references in the literature to instances of a water pump installed on the fourth deck of a large diesel ship that could be heard in the staterooms of the second deck because of the resonance of the water column in the pipes. A change in the length of the water pipe made it possible to eliminate the resonance and reduce the noise by 26 decibels (135). In another but similar case the noise produced by a hydraulic pump was audible four decks above the pump also because of resonance phenomena in the hydraulic line. The noise was reduced when a filter was installed in the line.

Noises travelling in liquid-carrying pipes can also be reduced to a large extent by the introduction of a device similar in principle to a ventilation system muffler and schematically outlined in Figure 180. This device consists of a sleeve containing a layer of material (soft foamy or porous material) whose acoustic resistance is considerably less than that of the water. A sealing rubber sheet is inserted to prevent the pores from filling up. An internal perforated pipe of thin sheet metal prevents vortex formation and disturbance of the flow where the silencer is installed.

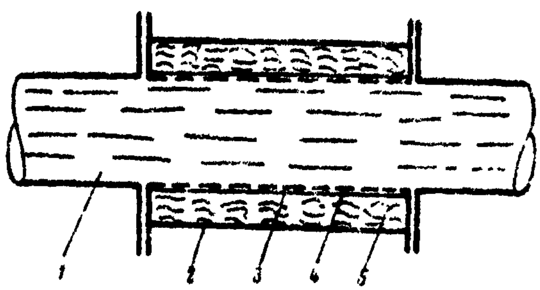


Figure 180. Arrangement of a silencer on a water pipe.
1 - the pipe; 2 - metal sleeve; 3 - perforated screen;
4 - rubber diaphragm; 5 - "penoplast" or "poroplast."

Such a silencer can eliminate the above-mentioned types of resonance but not the vibrations propagating along the pipe walls. This requires additional sound isolating joints or gaskets between the pipe flanges; the usual methods of vibration absorption are considerably

less effective in the case of piping. This is due to the thickness of pipe walls and the considerable intensity of the longitudinal vibrations in the walls, which cannot be absorbed by damping coatings. The installation on piping of vibration-inhibiting masses in the form of solid circular sectors (wedges) gives good results. These sector wedges can be combined with the expansion joints in the piping (133).

The noises produced by the elevators and hoisting equipment are made up of the noises of their machinery and their movements. The reduction in the physiological effect of the latter noise, and its localization, can be achieved by separating the elevator shafts from the hull structure by sound isolating mounts and the introduction of rubberized guide rollers moving up and down the shaft. To eliminate any sharp noises that are conducted by metal structures, the elevator door locks should be equipped with rubber inserts and the internal surface of the elevator trunk should be coated with sound absorbing material. For means of reducing the noises produced by hoisting machinery at its source as well as machinery used for sanitary and daily purposes, see Chapter 19.

59. Acoustic Requirements in the Arrangement of Compartments and Machinery

One of the practical methods of noise abatement is a proper arrangement of compartments having different noise levels, the entrances to these compartments and the distribution of noise-producing objects in them.

Instances of the wrong layout of ship's quarters are cited in Figure 181. In Figure 181-a, the compartment containing noise auxiliary diesel generators is located above the living quarters; moreover, the wall of one of the staterooms adjoins the freight elevator trunk.

In Figure 181-b (plan), the hatch from the machinery space into the corridor is directly next to the staterooms and opposite the entrance to the sick bay. For acoustic reasons, this hatch should be moved to the spot indicated by the dotted lines.

In Figure 181-c, the foundation of a noisy machine is located next to a water tank which, in turn, adjoins the wall of a passenger lounge. Obviously, the sonic vibrations from the foundation will travel through the liquid to the wall of the lounge causing noise in it. This noise could be reduced if the machine were removed from the vicinity of the tank and placed on efficient vibration isolation mounts, and the foundation made fairly massive.

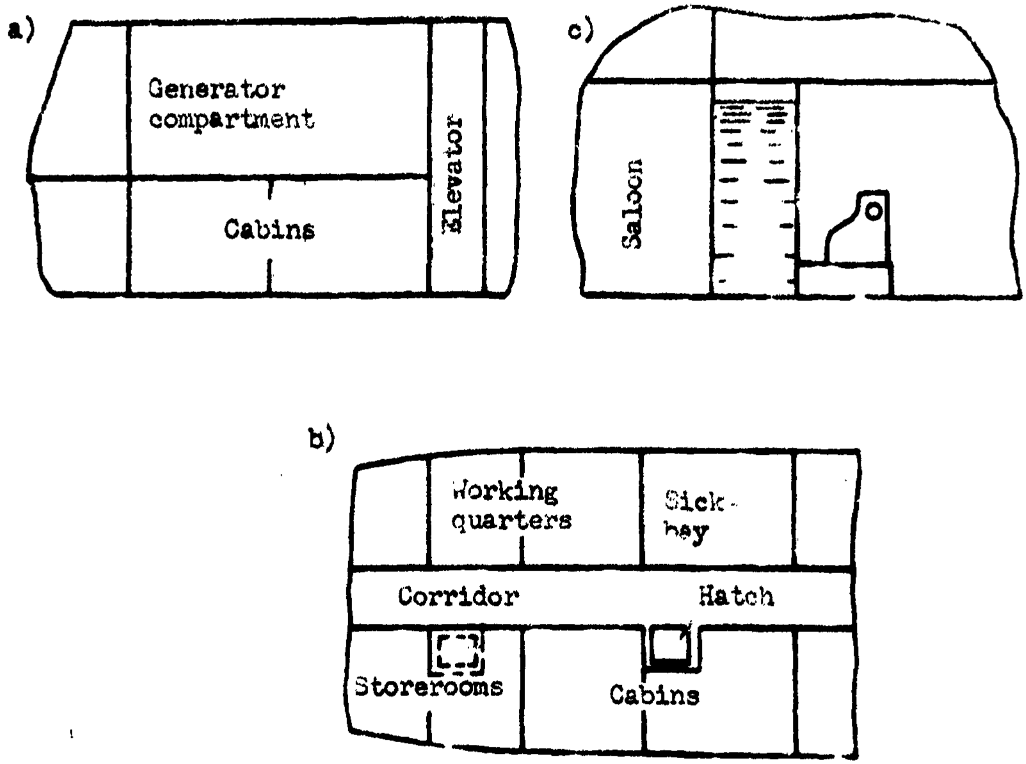


Figure 181. An irrational arrangement of the ship's quarters and the machinery in them, from an acoustical point of view.

When planning the layout of the ship's quarters and their distribution throughout the ship, account should be taken also of the sound isolating nature of passageways, cofferdams and similar seldom occupied quarters (storerooms, etc.) which could be located between noisy and quiet compartments.

60. The Role of Acoustics in the Designing, Construction and Operation of a Ship

The problem of acoustics is frequently raised toward the end of the designing and even the construction of a ship. The result is the construction of ships with a high noise level. Thus an 18,000-ton turbine tanker built by the Weser plant in Bremen in 1958 was rejected by the customer because of the high noise level of the main reduction gearing. Two years later the ship was sold for two-thirds of its construction cost. But when the modern sound-control measures are made

use of during the construction of ships, their noise level can be substantially reduced without special cost.

One of such measures, maintaining the acoustical point of view in the planning of ship layouts, was mentioned in a previous paragraph. Other examples of the role of acoustics include: the selection of machinery and systems for their acoustic characteristics; rigid noise reduction demands on the plants producing the machinery, and the allocation of part of the ship's useful load for noise-abating treatments. The designers of ships, particularly passenger ships, must devote as much attention to all these problems as to the speed, strength and operational reliability of the ships.

The design bureaus connected with the shipyards and shipbuilding plants, and the laboratories of these plants are fully equipped for simple acoustical investigations on ships. Such work designed to improve the acoustic characteristics of the ships should include particularly: the discovery of the noisiest machines and the most intensive components of their noise spectra (if such information is not furnished by the manufacturers); the determination of the extent of muffling required in the air ducts; investigation of the acoustic effect of vibration isolation mounts; definition of the areas where the noise level is determined by sonic vibration and where by the airborne noise of the machines; the design of resonance sound absorbers; the investigation of the resonance characteristics of the ship's hull structure.

Acoustic control should be maintained during the construction of the ship, especially in connection with important installations (vibration isolation mounts, double sound isolating bulkheads) which will eventually be difficult of access. Highly desirable also is the introduction of acoustic control in the plants producing mechanical equipment (for the methods of such a control see Chapter 20).

The negligent and improper operation of the ship's machinery can also raise their noise level. A soundproof structure may in time reveal cracks, rigid cross-connections, and hardened elastic inserts deteriorating the acoustic properties. It would therefore be quite useful to control the proper operation of the machinery and make periodic checkups (at least during ship repairs) of the soundproof efficiency of the ship's compartments. Whenever a ship is modernized, attention should be paid to the possibility of introducing new noise reducing improvements.

The proper organization of work in ships' acoustics during the planning, construction, testing, operation and repair of ships, and the dissemination of acoustic knowledge among a large circle of

shipbuilders would considerably reduce the noise level in this type of transport. It would also eliminate such paradoxes as shipbuilders themselves preferring to travel on the few remaining old quiet steamboats still plying the rivers rather than on their own diesel ships.

PART III

CONTROL OF NOISE AND SONIC VIBRATION

IN SHIPS' MACHINERY AND EQUIPMENT AT THE SOURCE

CHAPTER 16

THE SOURCES OF THE NOISE OF SHIPS' MACHINERY

61. The Origin of Noise in Machines

The problems involved in the struggle with the noise of machines and mechanisms at the source are a part of a very complicated and as yet inadequately explored area of technical acoustics.

The noise of machines is originated by the elastic vibrations both of the machine as a whole and of its separate parts. These vibrations in various shipboard machines are produced by mechanical, hydrodynamic and electrical phenomena which are determined by the design and nature of operation of the machine, technological inaccuracies in its construction and, finally, the conditions of operation. Noises may be divided into those of mechanical, aerodynamic and electromagnetic origin.

The noises of a mechanical origin are produced by the following factors: the inertial exciting forces arising from the movements of machine parts at variable accelerations; the impact of moving parts at articulated joints due to the unavoidable clearances; friction in the joints of machine parts.

Many shipboard machines are subjected to vibrations by oscillations of the working medium as a result of the hydrodynamic and aerodynamic processes occurring within the machine.

These aero-hydrodynamic sources of noise include: vortical processes occurring in the working medium; vibrations produced by the rotation of

laded rotors; pulsations of pressure in the working medium, oscillations within the medium, occasioned by non-uniformity of the flow into a rotor. Cavitational processes are another source of noise in hydraulic mechanisms.

Noises of an electromagnetic origin are produced in electric machines and equipment. These noises are caused primarily by the interaction of ferromagnetic masses under the influence of magnetic fields which vary in both time and space.

Below is a brief discussion of the basic physical processes causing noises in machines. A more detailed discussion of the sources of noise which are characteristic of various shipboard items of machinery, and certain methods of reducing that noise, will appear in the following chapters.

6?. Sources of Noise of a Mechanical Origin

Any mechanism in operation is to some extent subjected to the effect of unbalanced inertial forces which cause it to vibrate.

Inertial forces are the major exciting forces causing the vibration of reciprocating engines. The formation of these forces is due to the uneven movement of the piston-related parts and the constructional imbalance of the connecting rod -- crank mechanism (in some designs of engines).

Disturbing centrifugal forces produced by unbalanced masses appear in the rotating parts of machines. There are usually some deviations from the prescribed geometric dimensions of these parts in the course of their production. The material from which the parts are made may be structurally inhomogeneous. Any two linked parts of a machine may have their centers displaced to some extent. All this is conducive to the appearance of unbalanced masses (the so-called unbalance of rotating parts).

There are two types of unbalance associated with rotating parts, static and dynamic. In the first, the distribution of the density of the material and technological inaccuracies are such that all the unbalanced masses can be reduced to a single mass of weight Q_H which is displaced in relation to the rotative axis of the part by an amount r (Figure 162). This is accompanied by a general displacement of the center of gravity of the rotor (or of any rotating part) equal to

$$e_c = \frac{Q_H \cdot r}{Q_p},$$

where Q_p is the weight of the rotor.

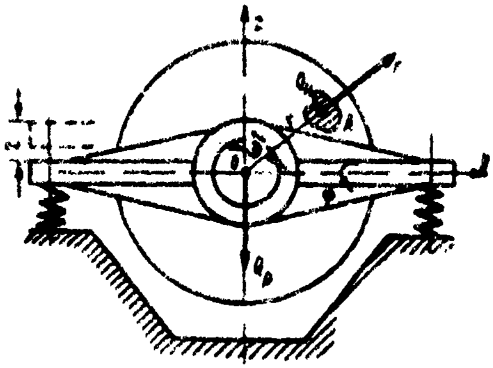


Figure 182. Vibration of a machine caused by the static imbalance of the rotor

The quantity ε_c is called the rotor eccentricity.

The rotation of a rigid rotor produces an unbalanced centrifugal force F which is proportional to the eccentricity ε_c:

$$F = m\varepsilon_c\omega^2, \qquad (234)$$

where m - represents the mass of the rotor.

ω - the angular speed of the rotor.

Force F acts in a plane perpendicular to the axis of rotation causing the machine to vibrate at the rotational frequency $f = \frac{\omega}{2\pi}$.

The vibration of the machine is greatly affected by the rotor elasticity. If the angular speed of the rotor is close to the frequency of its free transverse vibrations, the dynamic deflection of the rotor is greatly increased, involving an additional increase in the unbalanced centrifugal force F. Actually, as is seen in Figure 183, the disturbing force in an elastic rotor increases in proportion to the dynamic deflection ρ:

$$F = m(\varepsilon_c + \rho)\omega^2. \qquad (235)$$

When the disturbing force frequency coincides with that of free rotor vibrations, the dynamic deflection of the rotor increases to its highest value and the vibration of the machine reaches its maximum. The angular speed ω_0, for this case, is called the critical speed. The increase in the amplitude of the rotor deflection, at resonance, is limited exclusively by the damping forces within the rotor and in its supports.

In the case of dynamic imbalance, all the unbalanced masses of the rotor are reduced to two mass weights, Q_{H1} and Q_{H2}, lying in different transverse planes I and II (Figure 184). The rotor when turning at an angular speed ω produces an unbalanced dynamic moment:

$$M_d = F_1 a + F_2 b = (Q_{H1} r_1 a + Q_{H2} r_2 b) \frac{\omega^2}{g}, \qquad (236)$$

where g - represents the acceleration of the force of gravity;

a and b - the distances from the center of gravity of the rotor, O, to the planes I and II;

r_1 and r_2 - the distances from the axis of rotation to the unbalanced masses Q_{H1} and Q_{H2}.

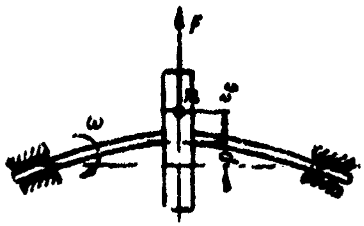


Figure 183. The effect of rotor deflection on the magnitude of the inertial exciting force F.

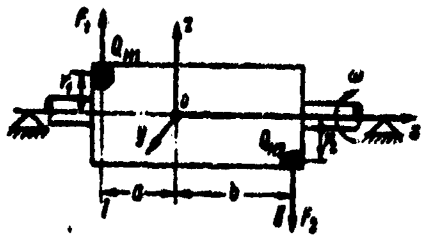


Figure 184. The exciting moment produced by dynamic imbalance of the rotor

The dynamic moment produces rotary vibrations of the machine in the zox and xoy planes (Figure 184). The vibration frequency caused by the dynamic unbalance also corresponds to the rotational frequency of the rotor f.

Actually the rotors of shipboard machinery are always characterized by both types of the unbalance. In this case the machine is affected by the exciting moment, as given by formula (236), and the exciting force

$$F = F_1 - F_2 = (Q_{H1} r_1 - Q_{H2} r_2) \frac{\omega^2}{g}. \qquad (237)$$

It is commonly believed that the harmonic vibrations of the machine are caused by the static and dynamic unbalances of the rotor. But this is true only when there is no clearance in the rotor bearings. Certain clearances in the bearings, however, are found in all actual machines, and these cause some vibratory motion of the journal in the bearing (Figure 185-a). The exciting force in this case is determined not only by the rotational angle of the rotor but also by the position

of the journal in the bearing. This type of motion produces higher harmonics in the vibration spectrum with frequencies representing multiples of the rotation frequency -- 2f, 3f, 4f, etc. The amplitudes of the higher harmonic components of the vibration are much lower than those of the basic harmonic; they are determined by the ratio of the rotor eccentricity ε_c to the magnitude of the clearance in the bearings s.

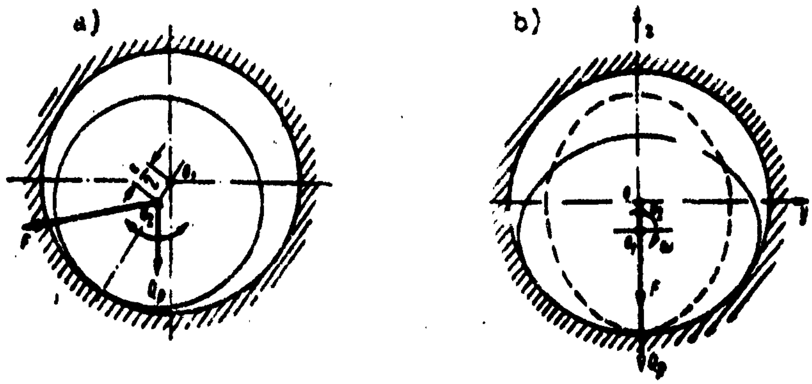


Figure 185. Rotor vibration due to (a) bearing clearance and (b) oval shape of shaft journal.

Another source of vibration is the ovality of the shaft journal, which causes a periodic displacement of the rotor's center of gravity. Its center of gravity is displaced twice during a single revolution -- from the lowermost position O_1 to the uppermost position O_2 (Figure 185-b). This produces an inertial disturbing force F which acts upon the body of the machine along the z axis. The vibration frequency produced by the ovality of the shaft journal is equal to twice the rotational frequency.

The improper installation of machinery is frequently the cause of intensive vibrations: inaccurate centering of shafts, sagging shafts, etc. It should be borne in mind that the elastic couplings used in shipboard machinery can only reduce the adverse effect of these factors, and cannot eliminate the effect entirely.

The vibration amplitude caused by the angular or parallel displacement of shafts (Figure 186-a, b) is proportional to the magnitude of such displacement, and the frequency is equal to the rotational frequency f. A change in the centering of the shafts connected by a coupling of the fingered type produces a vibration frequency equal to

$$f_k = fz_k \qquad (238)$$

where z_k represents the number of fingers (pins) of the coupling.

An initial warp of the shaft (Figure 186-c) also causes vibration.

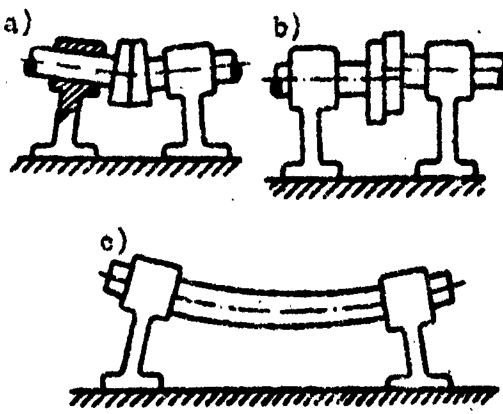


Figure 186. Installation errors which cause vibration of machinery.

The second of the above-mentioned chief causes of vibrations is met with fairly often. Basically it consists of the periodic and nonperiodic impacting together of moving parts.

Many shipboard machines are equipped with a power gear train. Any lack of precision of the gear manufacturing process produces a distinct impact between the engaging teeth and causes intensive vibrations. The vibration frequency in this case is determined by the frequency of the recurrence of the inaccuracies and the number of revolutions.

One of the major sources of vibrations in piston machines is the impact of the piston against the cylinder liner during its cross-over at upper and lower dead centers. In this case the vibration intensity is determined by the size of the clearance and increases with increased clearance. The vibration frequency corresponds to that of free vibrations of the walls and covers of the cylinder block. The same thing happens in the crank-piston rod assembly (the impacts of the piston against the upper end of the connecting rod, impact of the crank of the lower end, etc.) and in the intake and exhaust valves.

Impacting parts are a source of vibration also in roller bearings. Increased clearances in the bearing produce a chaotic movement of the steel balls within the clearances. The clashes of the steel balls against the rings and separator cause these parts to vibrate at their natural frequencies. In direct-current electrical machines the impacts of accumulator brushes against the plates are a source of vibration.

Let us examine the third cause of mechanical vibrations, the friction of rubbing parts. The most intensive vibration is caused by dry friction. The use of a lubricant reduces the forces of friction as well as decreasing the vibration. Forced lubrication creates a situation wherein the rubbing parts are separated by a layer of lubricant. In that case dry friction is completely eliminated and the vibration is reduced to a minimum.

Poorly finished shaft journals and bearing bushings as well as inadequate lubrication may produce vibration due to the action of a constant force (so-called autovibration). Similar self-excited vibration of the separators occurs in roller bearings due to the different diameters of the balls and their resultant varying friction against the separator.

63. The Sources of Noise of An Aero- and Hydrodynamic Origin

In hydraulic mechanisms (pumps, blowers), the rotating bladed wheels produce periodic pressure pulsations. At any point of their swept area, the blades enter the working medium differently, causing an alternating rarification and condensation in the flow. (Figure 187). The vibration produced within the medium in this process is called the "rotational sound."

One of the most intensive aerodynamic sources of noise is vortex formation in the flow inside the machine. A vortical noise is produced by the movement of a solid body in a gaseous medium, when a body is in a flow of working medium, or when the medium exits from a nozzle. Vortices and vortical noises are produced by a change of pressure in the working medium (Figure 187). The intensity of a vortical noise is determined by the size of the body and the velocity of the onrushing flow. Vortex noise has a continuous spectrum occupying a wide range of sonic frequencies.

Vortex noise can travel along the working fluid into the atmosphere as well as into the walls of the machine; the latter in this case produces sonic vibrations in other parts of the machine as well as airborne noise. A coincidence of the frequency of break-away of vortices with that of the free vibrations of the body may produce intensive resonance vibrations of that body.

A source of intensive noise in ships' pumps is the cavitation formed on the blade surfaces when the peripheral speed is high and the inlet pressure low. Cavitation noise, as a rule, appears even in the early stages of cavitation. Even the smallest source of cavitation which hardly affects the efficiency of the pump is sufficient to produce intensive noise.

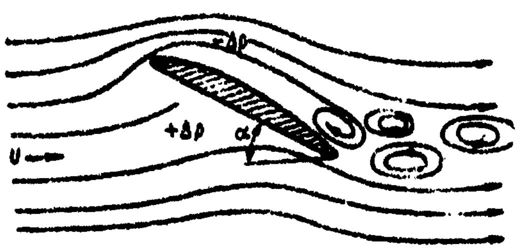


Figure 187. The creation of vortices and the "rotational sound" when a solid body is placed in a fluid flow.

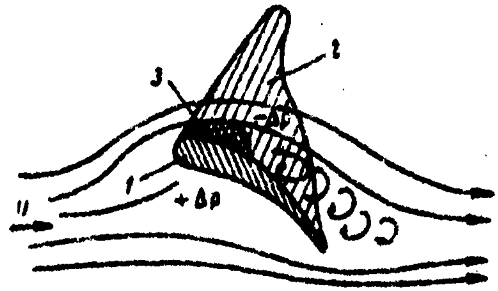


Figure 188. Cavitation on the rotor blade of a pump

1 - blade
2 - the distribution of negative pressure on the blade
3 - cavitation region

Cavitation is formed in a fluid flow (Figure 188) at points where the rarification process produces a break in the fluid's continuity. A pocket is thus formed and filled by the air which is dissolved in water, and in cases of intense rarification, by water vapor. When the vapor or gas bubble subsequently closes (due to vapor condensation or dissolving of the gas), the water particles dash toward its center, thereby sharply raising the pressure. This sudden increase in pressure is nothing more nor less than a sound impulse. Like any impulse, it consists of a number of components with different frequencies, mostly in the range of medium and high sonic frequencies. Cavitation noise produces a very strong effect on the sense of hearing in view of its numerous high-frequency components.

A similar phenomenon is found in water pipes at elbows and valves where, due to break-away of the flow, cavitation sources are also formed.

64. The Sources of Noise of Electromagnetic Origin

One of the major sources of noise in electric machines is the stator vibration caused by alternating magnetic fields ("stator hum", "magnetic noise").

Let us examine the formation of exciting forces produced by alternating magnetic fields in the case of a direct-current electric motor or generator. It is known that between the poles and the armature of an electric motor there are magnetic attractive forces which are defined by the following formula

$$F = \nu b_\nu^2,$$

where b_ν — represents the instantaneous value of the magnetic field induction;

ν — a coefficient of proportionality.

The intensity of magnetic induction b_ν is a function of the size of the gap between the armature and the poles of the motor. As the armature rotates through point O_1 for example, the winding grooves on its surface produce a constant, periodic change in the size of the gap (from δ_1 to δ_2) (Figure 189). A full cycle of changes in the magnitude of the gap occurs during the length of time it takes the armature to turn a distance equal to the pitch of the grooves (or of the teeth) (from position 1 to position 2). Obviously, the change in the magnetic induction b and the magnetic force F occurs at the same frequency and according to the same rule.

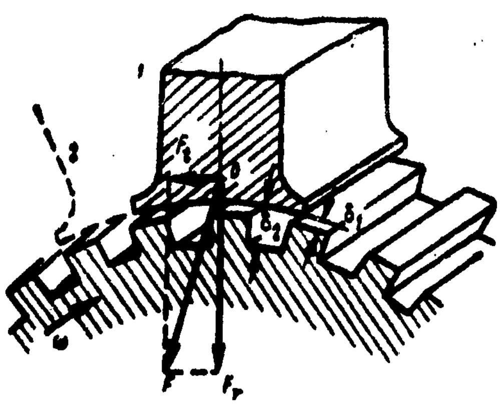


Figure 189. The cause of exciting forces in a direct-current electric motor.

In general, the frequency of magnetic vibrations and noise is defined by the following formula

$$f_M = f z_3 i, \qquad (239)$$

where f - represents the rotational frequency;

z_3 - the number of armature grooves;

i - 1, 2, 3, . . .

The disturbing force F may be divided into a radial component F_r and a tangential component F_t. Both the radial and tangential components produce a vibration of the stator yoke. The vibration intensity can increase markedly when one of the natural frequencies of the stator vibration coincides with the fundamental frequency of the disturbing forces F_r or F_t or of their harmonics.

Similar vibrations occur also in alternating-current machines. As is known (112), a number of higher harmonic fields may be produced in the air gap of an asynchronous motor in addition to the main magnetic field. These fields include:

(a) higher harmonic fields of the windings, produced by the non-sinusoidal distribution of the magneto-motive force in the air gap;

(b) fields in the teeth, produced by the alternating magnetic conductivity in the air gap of the machine;

(c) higher harmonic fields arising from various asymmetries in the magnetic circuit of the machine;

(d) higher harmonic fields arising from the asymmetry of the voltage supplied to the machine.

Experimental investigation has revealed that the radial components of the tooth and winding fields play a major part in the formation of magnetic noise in electrical machines.

Insufficiently tightly packed motor and transformer plates can be subjected to a periodic "loosening" under the action of alternating magnetic fields. This sometimes causes a very intensive noise.

Theoretically, sonic vibrations can be produced also by magnetostriction, that is by the changing of the form and dimensions of ferromagnetic plates under the action of periodic magnetic fields. But the change in dimensions of active iron in ordinary electric machines does not usually reach large values, and the noise of magnetostriction does not reach the level produced by the attraction of magnetic masses.

65. General Principles for Controlling the Noise and Sonic Vibration of Machinery and Equipment at the Source

The following design measures are now being used to reduce the sonic vibration and airborne noise of shipboard machinery:

(a) a reduction in the energy of the disturbing forces or a redistribution of them in time;

(b) a separation of the natural vibration frequencies of the parts of the machine from those of the disturbing forces;

(c) the use of vibration- and sound-isolating treatments in the structure of the machine;

(d) an increase in the scattering of vibrational energy (vibration-damping and sound-absorbtion.

As a rule, it is impossible to reduce the noise to any considerable extent by using only one of the above-mentioned means. Effective control of the vibration and noise of machinery calls for the use of all possible means of noise reduction.

The disturbing forces can also be reduced in the design stage of the machine by developing an appropriate design or by selecting an operating regime that would meet the acoustic requirements.

The measures during design, which reduce noise, include: a change in the shape of the blades in blowers; a reduction in the dimensions of roller bearings; the substitution of sliding bearings for ball bearings; a reduction in the ratio of the masses of the rotating parts to the fixed parts of the machine; tapering the grooves of the rotor in electric motors, etc. A reduction in the number of revolutions of rotors and of blower and pump blades will always reduce the exciting forces. Another effective method of controlling the noise of machinery and equipment is by reducing the velocity of the working medium.

The disturbing inertial forces and the vibration produced by imbalance, roller bearings, oval-shaped journals, poorly centered shafts, etc. depend primarily on the precision of manufacture of the various parts and the quality of the installation job. An improvement in production technology for machines and mechanisms is an effective means of controlling such vibrations.

The reduction of vibrations produced by the friction of the different parts upon each other depends largely on the proper operation of the machine. The timely and proper lubrication of all the rubbing parts of the machine will reduce vibration and noise.

If the noise of a machine is caused by intensive resonance vibration of its parts, the natural vibration frequency of each part can be tuned away from the frequency of the disturbing force acting on that part. This can be achieved by changing the stiffness of the parts (increasing or reducing their thickness, additional stiffeners, the replacement of flat-surface parts by nonflat ones, etc.), as well as by the introduction of additional balancing masses.

It is known from the theory of vibrations that the vibration amplitude in the region of resonance is to a considerable extent determined by the internal damping quality of the material from which the part is made. The use of materials with high damping characteristics makes it possible to control the vibration and noise of machines by scattering the vibration energy within the material. In some cases it is possible to introduce absorbers of airborne noise into the structure of the machine.

The reduction of the vibration and airborne noise of machines can be achieved by introducing vibration- and sound-isolating treatments into their structure. Such measures reduce the transmission of disturbing forces to the external surfaces of machines, thereby reducing the radiated noise.

CHAPTER 17

CONTROLLING THE NOISE OF REDUCTION GEARING AND MARINE INTERNAL COMBUSTION ENGINES

66. Characteristics of the Noise of Gear Transmissions

The major source of noise in marine geared turbine drives is the reduction gears. The airborne noise of the gearing is produced by the natural and forced vibrations of the gear wheels and parts of the reduction gear structure.

The vibrations predominant in gear transmissions are caused primarily by the contact engagement of the gear and pinion wheels. The frequency of these vibrations is equal to

$$f_{K3} = f\, z_t\, i \tag{240}$$

where z_t - is the number of teeth on the gear wheel (pinion);
f - the number of revolutions of the wheel per second;
i - 1, 2, 3, ... - - the number of the harmonic of the vibration.

The most important quantity in the noise spectrum is the first harmonic ($i = 1$).

The vibrations produced by the accumulated errors in the circular pitch of the pinions and gear wheels may play an important part. In that case the frequency is determined by the following formula

$$f_e = fi \tag{241}$$

The formation of uneven areas on the teeth during machining (Figure 190) may also produce vibrations. The defective performance of the worm drive of the gear-cutting machine (irregular pitch of the worm wheel, play in the worm) produces parallel sections of raised bumps and transitional areas ("waves") on the tooth surfaces along the contact

line. The peripheral distances between the lines of unevenness correspond to the pitch of the dividing gear in the gear train of the cutting machine, and the vibration frequency caused thereby is therefore determined by z_d, the number of teeth in the dividing gears of the gear-cutting machine.

$$f_d = f z_d i \,. \tag{242}$$

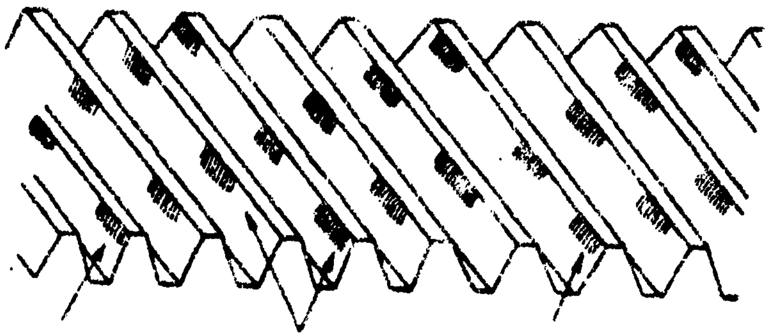


Figure 190. Rough spots on the surface of gear teeth (indicated by arrows).

As before, the first harmonic ($i = 1$) plays a major part but it too has a fairly high frequency.

Occurring simultaneously are the vibrations of the gear wheels at frequencies equal to the natural frequency of the individual teeth or of the wheel as a whole.

Example 57. Determine the frequency of the major components of the airborne noise spectrum of a ship's reduction gear with a capacity of 3,000 hp and with n = 4,550 rpm. The diagram of the reduction gear and the necessary kinematic data are given in Figure 191. The gear wheels are cut by a machine whose dividing gears have the following number of teeth: z_d = 120; 360 and 600. The number of teeth on the step I (first reduction) pinion wheel is z_I = 35, and on the step II pinion wheel z_{II} = 250. The number of propeller shaft revolutions is n_p = 180 rpm.

Solution. The rotational frequency of a high pressure turbine pinion is

$$f_I = \frac{n}{60} = \frac{4{,}550}{60} = 75.7 \text{ cps.}$$

The frequency of the contact engagement of the gear and pinion of step I is:

$$f_{K3} = f_I z_I i = 75.7 \cdot 35 \cdot i = 2650\ i\ \text{cps}.$$

The rotational frequency of the propeller shaft (bull gear) is:

$$f_{II} = \frac{n_p}{60} = \frac{180}{60} = 30\ \text{cps}.$$

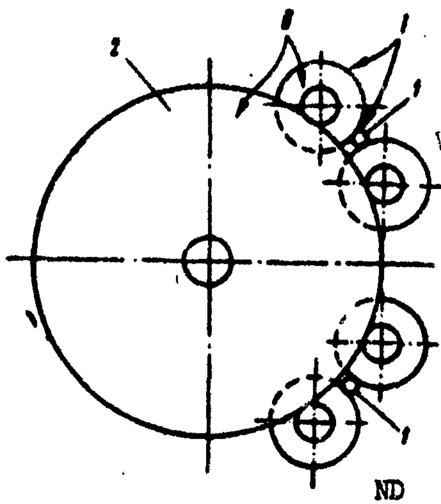


Figure 191. Diagram of a ship's reduction gear for a compound turbine (see example 57).

I - first reduction; II - second reduction; 1 - first reduction pinion; 2 - bull gear.

The frequency of contact engagement of the bull gear and the Step II pinion is:

$$f_{K3} = f_{II}\ z_{II}\ i = 3 \cdot 250 \cdot i = 750\ i\ \text{cps}.$$

The frequencies produced by the unevennesses on the tooth surfaces of the step II gear, with $z_d = 120$, are

$$f_d = f_{II}\ z_d\ i = 3 \cdot 120 \cdot i = 360\ i\ \text{cps}.$$

By analogous computations the frequency of cyclical errors produced by unevennesses on the tooth surfaces when

$z_d = 360$ ($f_d = 1{,}080i$ cps) and

$z_d = 600$ ($f_d = 1{,}800i$ cps),

can be found.

The presence of a considerable number of disturbing forces of an impact nature in the reduction gearing (as may be seen from the above example) is responsible for the intensive resonance vibrations of the parts of the reduction gear (torsional, transverse and axial vibrations of the gear wheels and pinions, and bending vibrations of the structure). As a result of this, marine reduction gears have a continuous noise spectrum covering a wide range of frequencies from several tens of cycles to several kilocycles. Frequencly distinguishable in the continuous noise

spectrum of such gearing are several components corresponding to the resonant as well as to the forced vibrations of its parts. Some typical reducing gear noise spectra which, according to H. Zink (172), are actually found in practice, are shown in Figure 192. The author divides these into six groups.

Group I (Figure 192-a). The spectrum is predominated by the contact engagement frequency f_{k3} and its harmonics $2f_{k3}$ and $3f_{k3}$ whose intensity is much lower than f_{k3}. Such spectra are produced by a predominant tooth engagement impulse coupled with a high degree of damping (for example, when there are large errors in the pitch, too small an amount of overlap, or an error in the angle of inclination of the tooth).

Group II. (Figure 192-b). The f_{k3} frequency of the teeth is hardly discernible but its $2f_{k3}$ or $3f_{k3}$ harmonics are prominent.

Numerous investigations have shown that in this case there are also intensive gear vibrations produced by impacts during tooth engagement.

The f_d component corresponds to the frequency of the dividing gear of the gear-cutting machine (the low frequency components in Figure 192-b are not characteristic, being determined by external noises in the compartment where the measurements were made).

Group III (Figure 192-c). A very large number of frequency components is noted in a wide area of the spectrum. This is brought about by impulses generated by numerous causes: tooth engagement, errors in engagement caused by heat treatments, etc.

Group IV (Figure 192-d). The f_d frequency prominent in the spectrum is produced by the errors of the dividing gear of the gear-cutting machine.

Group V (Figure 192-e). Characteristic of this spectrum is the presence of numerous frequencies: f_1 and f_2 are unbalance frequencies and their harmonics, f_{k3} engagement frequencies as well as a variety of frequency differences. Such spectra may be produced by the unbalance of a pinion and coupling, engagement errors, etc.

Group VI. A spectrum from relatively low-noise gears in which the frequency of the discreet components produced by the reduction gearing is hardly noticeable (this spectrum is not shown in Figure 192).

The various methods of noise reduction (Table 25) should be applied according to the type of spectrum.

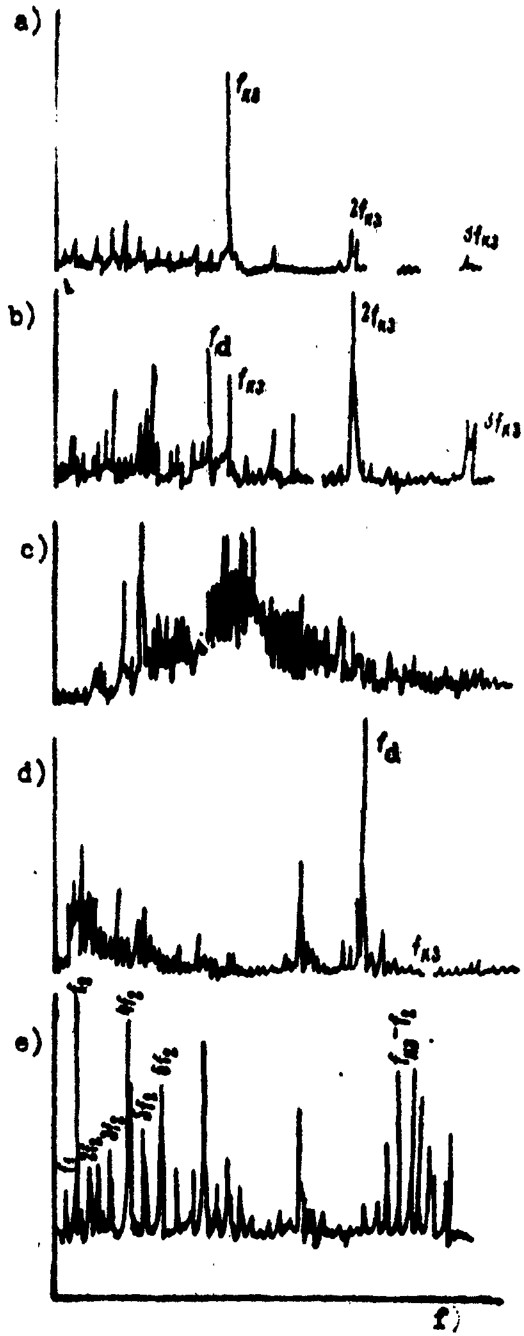


Figure 192. Typical spectra of the noise of geared drives.

67. Methods of Reducing the Noise of Marine Reduction Gearing and the Gear Transmissions of Machines

The noise level of geared transmissions is a function of the peripheral velocity of the gears, the dimensions of the gears and the quality of manufacture. At a distance of one meter from the housing, the noise level of a reduction gear, depending on its size, can be evaluated by the following expression (84).

$$\beta = 10 \log 6.6 S a^3 f^3 + 20 \text{ db} \qquad (243)$$

where S - effective radiating surface of the vibrating gear wheel, which depends on its dimensions and form of vibrations;

a - vibration amplitude of the gear wheel;

f - vibration frequency.

It may be concluded from relation (243) that in the design of reduction gears, one should strive for a limitation of the rotational speed of the gear and a reduction in its size. This can be achieved by the use of double reduction or 2-step gearing. At constant transmitted horsepower, reducing the rotational speed of the wheels produces a greater quieting effect than reducing the load on the teeth.

A decrease in the gear wheel module and an increase in the number of teeth within permissible limits will also reduce the noise because of the smaller size and lower peripheral speed of the gears.

Special types of gears -- globoid, helicoid, double herringbone -- are as a rule less noisy than straight tooth gears.

The chief means of reducing the vibration and noise for any given type and power of gear is an improvement in machining precision. According to (84), the permissible error in the pitch from the point of view of noise, as a function of the peripheral speed of the gear, should not exceed the following tabulated values:

U, m/sec	100	50	20	10	5	2
Δ, microns ...	16	20	35	70	140	350

The Δ values were calculated in such a way as to satisfy the requirement that the gear noise not exceed 70 decibels at a distance of one meter. These Δ values coincide well with practical data on the gear-cutting precision required for the production of low-noise gearing.

It has been established that increasing the number of teeth on the dividing gears of the gear-cutting machine reduces the magnitude of cyclical errors in the gear being cut. The machines currently in use therefore have a large number of teeth on the dividing gears.

A correcting device designed by the TsNIITMash (Central Scientific Research Institute of Technology and Machine Building, Moscow) is now being used to reduce the errors in gear cutting due to the machine's dividing gear train. The rough spots on the teeth produced by errors of in the dividing gear train of the gear-cutting machine are removed by buffing and scraping the teeth as well as polishing the teeth with a paste provided by the State Optical Institute (GOI).

Another important factor, determining to a great extent the vibration and noise intensity in the band of low and medium frequencies, is the balancing of the pinions and gear wheels of reduction gearing, especially the fast revolving pinions and the rotors of the turbines. A careful dynamic balancing of the turbine rotor, the coupling and the pinions, as well as the use of the TsNIITMash correcting device, has made it possible to reduce the reduction gear noise of a ship's turbo-generator from 106 to 96 decibels (68).

The reduction-gear noise of turbines is also affected by the nature of the lubricant as well as by the design of the non-toothed portions of the gear wheel. Thus by increasing the stiffness of attachment of the the wheel rim to the hub, the noise level can be reduced by 6-8 decibels.

Gear vibration, and consequently the noise it produces, can be reduced by the use of various frictional dampers. Rubber disks pressed on to the axially facing surfaces of a gear wheel can serve as such dampers.

A reduction in the noise of reduction gearing and the elimination of the high-frequency components of the noise can be achieved by manufacturing the gear wheels from materials with greater damping characteristics than ordinary structural steel. Plastic materials have recently been used on an increasing scale for the production of reduction-gear parts. Efforts are being made to use plastics for the production of pinions, as inserts between the gear rim and the wheel hub, for various parts of the housing, etc. Merely the replacement of the gearing's steel covers by a plastic cover (polyester-saturated fiber glass) reduces the noise by 2-8 decibels, and at high frequencies, up to 15 decibels (Figure 193).

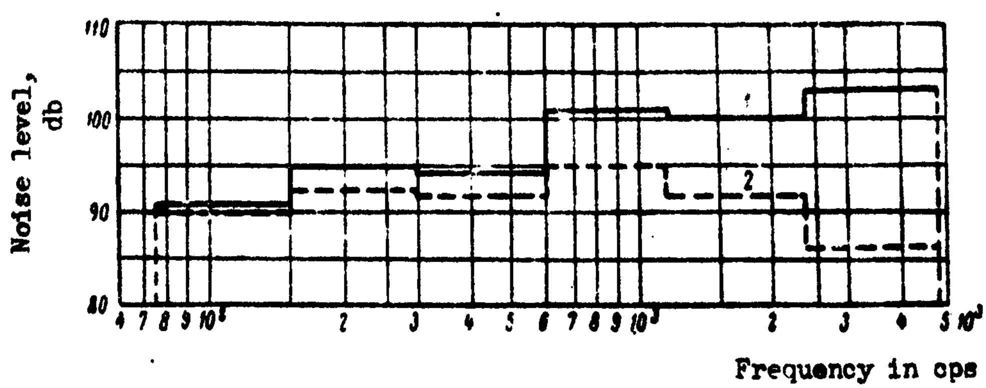


Figure 193. The effect of the material of the gear cover on its noise spectrum

1 - Steel cover 2 - Plastic cover

Another method is to use double-walled gear housings and fill the space between the walls with vibration-damping material such as bitumen. Successful experiments were carried out by the U. S. Navy in covering reduction gear housings with a vibration-absorbing coating (163). The gearing can be surrounded with additional sound isolating housings having material for absorbing airborne sound on their inner surfaces.

To reduce the vibration transmitted to the foundation and through it to the other ship's compartments, it would be worthwhile to insert vibration-isolating couplings between the shafts of the various units and place the entire geared turbine assembly on vibration isolation mounts.

The selection of any particular method of reducing reduction gear noise should be determined after the first production unit has been tested and its noise spectrum studied. Some of the recommended methods, corresponding to the nature of the noise spectrum, are listed in Table 25.

TABLE 25

PROCEDURES FOR REDUCING REDUCTION GEAR NOISE

Nature of spectrum	Major source of noise	Recommended noise reduction procedure
Group 1	Errors in pitch and angle of inclination of the teeth	Polish the teeth surfaces. Shift the phase of the accumulated error (in herring-bone gears)
Group 2	Large errors in pitch, caused by worn gear cutters in the gear cutting machine	Re-cut the gears and pinions with new milling cutters. Cover the gear with rubber facing.
Group 3	Pitch errors and random errors produced in the process of heat treatment.	Polish the teeth surfaces
Group 4	Rough spots on the teeth produced by errors in the dividing gears of the gear-cutting machine	Polish the teeth surfaces; scrape the rough spots; apply polishing paste. Recut the gear wheels with a larger number of teeth in the dividing gear.
Group 5	Pitch errors. Unbalanced pinions, gears and couplings. Radial and axial pulsation of the toothed part of the gear wheel relative to the journal.	Polish the teeth surfaces. Carry out dynamic balancing. Use greater precision in manufacturing gear and pinion blanks.

68. The Noises of Reciprocating Machinery

Internal combustion engines and particularly diesel engines are the noisiest of all the machines on shipboard. With respect to their origin, the noises of such engines can be classified as aerodynamic and mechanical.

The noises of an aerodynamic (or gasodynamic) origin include those produced by the inlet and exhaust systems of engines, various scavenging air pumps blowers and superchargers, as well as the noises produced by fuel combustion in the cylinders.

In some cases the suction noises produced by blowers and superchargers reaches a level of 115-118 decibels. The noise level produced by the suction strokes of an engine even without a blower or supercharger can be considerable. The frequencies of the most intensive components of suction noise in an engine without a supercharger is determined by the following formula:

$$f_s = \frac{f z_c k i}{z_k} \tag{244}$$

where z_c is the number of cylinders in the block;

z_k the number of simultaneously operating cylinders in one block;

k the number of impulses of one cylinder per revolution

($k = 1/2$ for a 4-cycle engine);

$i = 1, 2, 3 \ldots$

The noise of rotor-type (Roots-type) blowers consists of vortex noise, the noise produced by the pulsations of the air column in the pipe, by rotational noise and obstacles in the air stream. The noise produced by a centrifugal blower is of a similar composition including a very intensive component corresponding to a "siren noise" (see formula 260). The high-frequency, intensive noise of centrifugal superchargers produces a very unpleasant physiological effect.

The next source of noise is the combustion process in the cylinders. The great complexity of this problem and the lack of sufficient experimental data are responsible for the fact that we still do not have a well established general theory of noise produced during the combustion of fuel. One of the major reasons for that noise is the rapidity of the increase of pressure in the process of combustion.

In a number of internal combustion engines, especially large, slow turning marine engines, combustion noise is much lower than the total noise level and is masked by other more powerful sources of a mechanical nature. The mechanical sources of noise in diesel engines include: the crank-connecting rod gear; the fuel injection system; the valve gear; the various auxiliary mechanisms installed on the engine and their drives (particularly geared drives).

As revealed by Soviet investigations (40), the most important noiseforming element in reciprocating engines are the impacts in the crank-connecting rod gear, particularly the knocking of the pistons against the cylinder liners during cross-over.

It is known that during each revolution of the crankshaft the piston shifts from one side to the other several times, moving in the plane of the connecting rod's motion. The gap between the piston and the cylinder liner causes the piston to take on a certain velocity in the transverse direction, impacting shortly thereafter against the wall of the cylinder. These knocks produce an intensive ibration of the cylinder walls at their natural frequency, resulting in a noise level which is 3-9 decibels higher than the noise coming from other sources in the engine.

The intensity of the noise produced by impacts of the pistons against the cylinder liners is determined by the following factors: the number of revolutions per minute, the magnitude of the clearances in the bearings, the weight of the piston and the connecting rod, the magnitude and nature of the forces acting upon the piston, the ratio of the crank radius to the length of the connecting rod, the material and the thickness of the cylinder block and cylinder heads, the number of cycles of the engine, the number of cylinders, and the lubricant viscosity.

Inasmuch as piston impact (piston "slap") is the major source of noise in most engines, the noise levels of engines can be determined by a single formula. For this purpose we use the empirical formula proposed by V. I. Zinchenko (40) to measure the noise level at a distance of one meter:

$$\beta = 15.8 \log F_s + B_1 \text{ db} \qquad (245)$$

where $F_s = Dn\sqrt{i}$ is the noise factor for an engine with compression ignition (D-cylinder diameter, i - number of cylinders, n - rpm);

$B_1 \approx 70$ decibels in the case of 2-cycle and 4-cycle engines with a cast iron block.

Formula (245) holds true in the interval $F_s = 150$ -- 850 meters per minute.

Example 58. Determine the noise of a marine diesel engine from the following data: cylinder diameter $D = 300$ mm, number of cylinders $i = 4$, number of revolutions $n = 300$ rpm.

Solution. The noise factor of the engine is:

$$F_s = Dn\sqrt{i} = 0.3 \cdot 300\sqrt{4} = 180 \text{ m/min.}$$

According to formula (245),

$$\beta = 15.8 \log F_s + 70 = 15.8 \log 180 + 70 \approx 106 \text{ db}$$

In addition to the piston impacts, the noise in internal combustion engines is also caused by the crank-connecting rod-piston linkage (crank and top-end bearings) and the effect of inertial forces on that linkage.

In some cases engine noise can be produced by the torsional vibrations of the crankshaft (118). In the case of torsional vibrations, the movement of the crankshaft is such that the main journals vibrate within the bearing clearances in different directions and impact against the bushings at the critical shaft frequency. The result is a low frequency roaring or rattling noise. Figure 194 shows the relationship between the total noise level of a 6-cylinder engine and the number of revolutions. When n = 1,280 rpm, intense torsional shaft vibrations are produced by the exciting forces of the 6th order of the rpm. This produces a sharp increase in the engine noise level.

Naturally, all engines are designed in such a way as to avoid critical regimes in the operating region of their revolutions. But many engines operate fairly close to a critical regime; in such cases the torsional shaft vibrations are an additional source of noise.

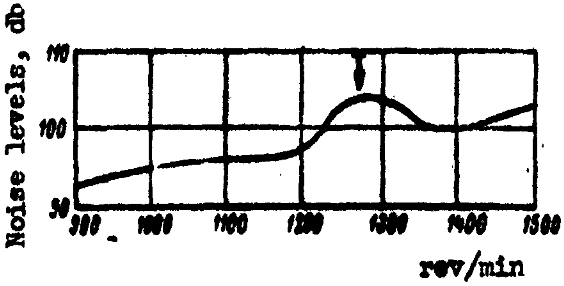


Figure 194. Effect of torsional vibration of the crankshaft upon engine noise. The arrow shows the critical speed.

The fuel apparatus can also be a source of noise in marine internal combustion engines due to the hydraulic and mechanical knocks (the lifting and the seating of the sprayer needles, the injection process, impacts upon the cams and the vibration of camshaft). The operation of the camshaft mechanism is accompanied by noise produced by the impact of the valves upon their seats, impacts on the cams, tooth contact noises in the driving pinions, etc. But as pointed out above, in most engines the major source of mechanical noise is the impacts of the piston at cross-over (dead centers).

69. <u>Factors to be Considered in Future Work on Noise Reduction in Marine Reciprocating Engines</u>

As has been mentioned, one of the very intense noises in internal combustion engines is produced by blowers and superchargers. The noise of reciprocating blowers and compressors can be reduced at the source by increasing the cross section of the flow-conducting elements and establishing a proper phase distribution. To reduce the unpleasant noises of the rotor-pinion (Roots-type) blowers, which have sharp tonal components, it would be worthwhile to install a reactive muffler tuned to the fundamental frequency and its basic harmonics at the blower inlet. The high frequency noise of centrifugal blowers can easily be quieted by active mufflers (see paragraph 43). Similar mufflers are also used to reduce exhaust noises.

The fight against the noise produced by the combustion process should be carried out by the following methods (40):

(a) an increase in the temperature of the cycle;

(b) replacing aluminum blocks with cast iron blocks, thereby usually reducing the noise by 5-6 decibels without changing the thickness of the cylinder walls;

(c) increasing the thickness of the cylinder walls in the vicinity of the combustion chamber;

(d) organizing the combustion process (including fuel treatment) in such a way as to prevent the appearance of shock waves in the cylinder, reducing the rate of increase of pressure and eliminating the sharp pressure change which occurs during the transition from the process of compression to the process of combustion.

The operation of an engine on the so-called M-process is an example of a rational working cycle from an acoustic point of view. In this case, as experiments show, the noise of the engine is hardly different from that which it produces when turned over by another power source (without combustion).

When an engine operates on the M-process, the fuel is injected by the nozzle in two directions into a spherical combustion chamber located in the piston. Most of the fuel enters the combustion chamber in the form of a jet, directed against the combustion chamber walls and covering a considerable portion of their surface with a thin film. About 5% of the fuel is atomized in the air and, bursting into flame, ignites the rest of the fuel, evaporating it from the chamber walls.

The reduction of the combustion noise of an engine using the M-process is achieved by gradual ignition (a "soft" combustion process), low injection pressure, and less complete fuel atomisation. A noise reduction is also facilitated by the fact that combustion occurs in a spherical chamber with thick walls. Besides, the trunk portion of the pistons is well lubricated with cooling oil. A layer of oil between the cylinder liner and the piston also reduces the intensity of the mechanical piston impacts against the cylinder walls at cross-over.

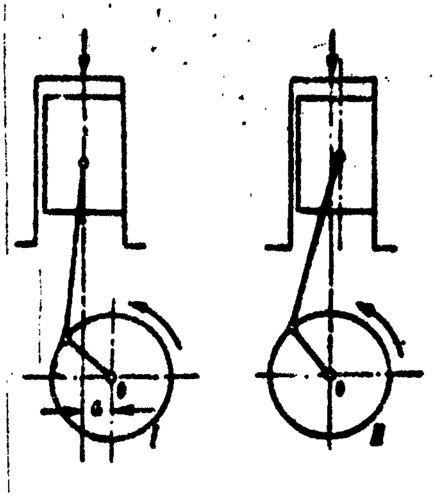


Figure 195. Displaced (I) and normal (II) crankgear. a - displacement of the axes.

In any type of working process, the intensity of the mechanical noise produced by the impacts of the pistons can be reduced by reducing the clearances, the piston-associated weights and the number of the engine's revolutions. For this purpose it is recommended to use a displaced crank-connecting rod-piston gear (Figure 195). The displacement of the cylinder axis $\bar{a}$ in relation to the crankshaft axis results in a redistribution of the normal pressure N and a reduction of it at cross-over. Moreover, the rate of change of the normal pressure $\frac{dN}{dt}$ is also reduced. The intensity of the impact noise is therefore less than in the case of ordinary crank gear.

It is best to displace the crank-piston rod gear by an amount equal to 20-30% of the crank radius. With such an arrangement the noise level is reduced by 5-7 decibels in comparison with engines having a normal crank-gear. Regardless of a rearrangement of the axes, the crankshaft of the engine should be well balanced.

The noise of the valve gear and the fuel injection system is considerably intensified by the vibrations of the cylinder block cover. This noise can be reduced by placing the covers on vibration isolating gaskets and fastening them with vibration-isolated bolts. Vibration- and sound-isolation of the fuel injection system also help to reduce the noise, as does the installation of hydraulic drives for the auxiliaries (especially in high-speed diesels). The replacement of the conventional-type valve

system by sleeve valves should produce good results. Some specialists (40) recommend that the development of a more powerful engine should be carried out not by increasing the number of revolutions or the cylinder diameter, but by supercharging and developing a better muffler for the supercharger inlet.

The most radical measure for reducing the noise of diesel engines on passenger ships is a reduction in the number of revolutions, that is a return to the slow turning diesel. As shown in chapter 5, the noise level of slow turning diesels is 14-16 decibels lower than that of high rpm diesels of the same horsepower. A further reduction in the noise is possible only by sound isolating and sound-absorbing treatments in the engine room itself (soundproof housing, partitioning, sound-absorbing coatings, local absorbers).

It should not be forgotten that steam engines are among the quietest marine engines. The noise produced by a steam engine is 20-25 decibels lower than that of a diesel of the same horsepower and is, as a rule, within the range of permissible norms (see Table 6). In certain types of ships, where noise is absolutely impermissible, the only logical type of engine to use would be the steam machine.

CHAPTER 18

THE NOISE OF ELECTRICAL MACHINERY AND METHODS OF REDUCING IT

70. Research on the Noises of Electrical Machinery

There are three types of noises in rotating electrical machines: magnetic noises caused by alternating magnetic fields, mechanical noises caused by rotor unbalance, vibrations in roller bearings, friction and impacts of the brushes, etc.; and airborne noises produced by the cooling fan and by the air flow in the passages of the machine.

A thorough investigation of electrical machinery noises under different operating conditions was made in the USSR as far back as the nineteen thirties (113).

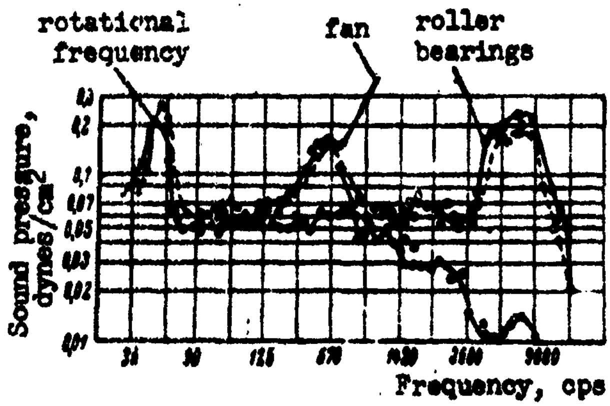


Figure 196. Noise spectra of an asynchronous motor (sound pressure in dynes/cm^2)

- - o - - o initial motor
- ● - ● - ventilator fan removed
— o — o — roller bearings replaced with plain bearings

An investigation of the noise is made by measuring the total noise level as well as the level of the spectral components. A comparison of the noise spectra obtained at the different regimes and operating conditions makes it possible to determine the causes of the noise and plan the methods of reducing it.

Cited in Figure 196 is an example of noise spectra produced by an alternating-current machine with a short circuited rotor ($n = 1$ kw, $n = 3,000$ rpm). In the 50-cycles per second frequency range, in this case, the noise is produced

primarily by an unbalanced rotor and by magnetic vibrations at the circuit frequency (50 cycles). The spectrum component at a frequency of 570 cps is the aerodynamic noise of the fan. In the band of 1,400-10,000 cps, the noise is produced by the roller bearings.

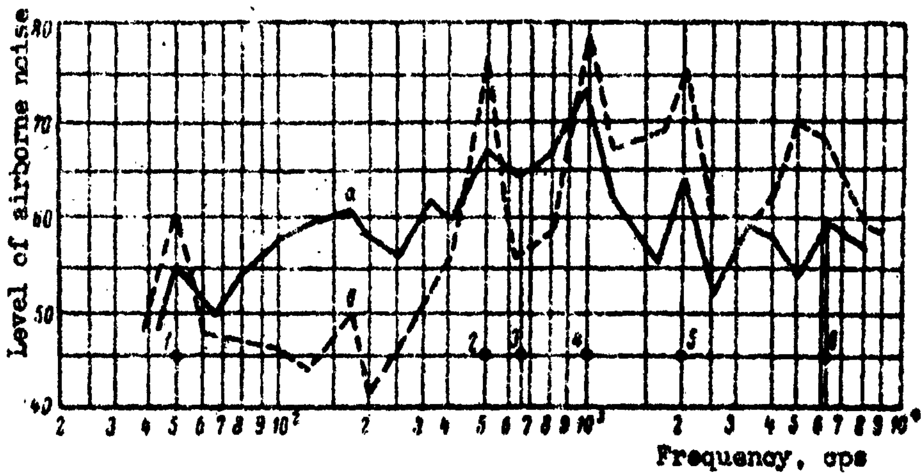


Figure 197. The spectrum of airborne noise and vibration of a ship's electric converter at n = 3000 rpm and N_e = 15.1 kw: a - airborne noise in db; b - vibration.

1 - A fundamental, caused by rotor imbalance; 2 - fundamental, due to ventilation noise from rotation of the rotor; 3 - fundamental of ventilation noise from the cooling blading; 4 - fundamental of magnetic noise of the generator; 5 - fundamental of magnetic noise of the motor; 6 - fundamental of brush noise of the motor.

The complete elimination of the ventilator fan noise in the case under consideration resulted in a reduction of noise intensity by only 2 decibels. But the replacement of the roller bearings by plain (sliding) bearings has reduced the motor noise by 7 decibels, which is a good achievement.

Figure 197 shows another example of the noise spectra and vibrations of a ship's electrical converter.

71. Reducing the Magnetic Noise of Electrical Machines and Transformers

It follows from the material outlined in paragraph 64 that the simplest method of reducing the magnetic noise is to reduce the magnetic induction in the clearances of the machine; for example, reducing the use of iron in the magnetic circuit or increasing the air clearance. This, however, involves an increase in the weight, size and cost of the machine.

A more rational arrangement would be skewed armature grooves. Skewing of the armature or rotor grooves facilitates a more uniform distribution of the magnetic flux in the air clearance and reduces the intensity of the teeth magnetic fields, thereby also significantly reducing the noise of the machine.

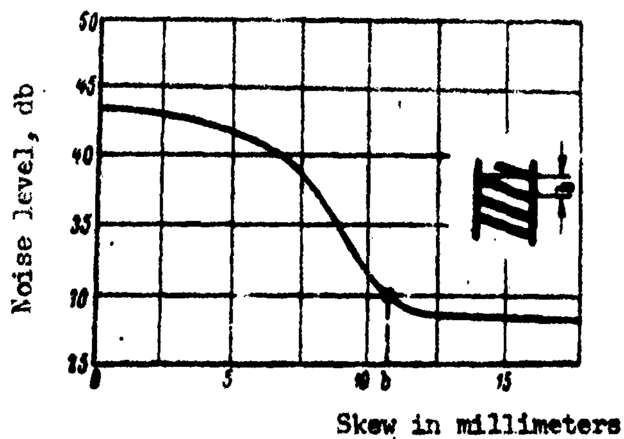


Figure 198. Relationship between the noise level and the skew of the grooves in an asynchronous electric motor (when idling).

Figure 198 shows the relationship between the noise level of an electric motor and the amount of groove skew (in an idling motor) (113). A considerable effect can be achieved by skewing the grooves of the rotor by one tooth width; a greater skew does not reduce the noise.

The magnetic noise of an engine can also be reduced by dividing the length of the armature into a number of electrically independent portions, out of line with each other. Thus produces a mutual compensation of parasitic magnetic fields within the machine. The use of semi-closed grooves, as well as special forms of the pole shoe designed to increase the interpole clearance around the shoe edges, can reduce the amplitude of the disturbing forces of the higher harmonic fields.

When developing new machines, care should be taken to avoid the coincidence of the disturbing force frequencies with those of the natural vibrations of the stator. The resonance can be tuned out by making the resonant part stiffer or changing its shape.

The method of isolating the disturbing forces from the stator yoke and the base frame has been successfully used on large turbogenerators as well as on small electric machines. For this purpose, the stator mandrels are fastened to the yoke by elastic steel springs, and the stator itself is attached to the frame by vibration-isolating rubber bushings (Figure 199).

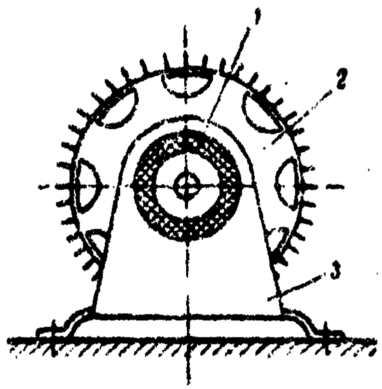


Figure 199. Isolating the vibration of an electric motor body from its base frame

1 - rubber ring; 2 - stator; 3 - base frame.

If the above-listed methods do not reduce the machine noise to any considerable extent, they should be supplemented by covering the stator with a damping coating. But this measure is effective only when the magnetic noise is caused by intensive resonant vibrations of the parts. By way of example, we should like to recall the case when a 30-mm thick damping coating produced a 16-decibel reduction in the resonant vibration of a large stator yoke (diameter about 1.5 meters) at a frequency of 1,100 cps (149).

A periodic compression of the laminated blocks occurs in transformers when the active iron is not adequately pressed. This is due to the periodic magnetic reversals which produce attracting and repelling forces. The fundamental frequency of the pulsations in the iron and of the concomitant noise equals twice the frequency of the circuit. There is also a number of more or less conspicuous higher harmonic components. Similar vibrations are observable in alternating-current motors and generators. To eliminate these, the cores must be pressed into great solidity and damped with some viscous material (bitumen, layers of bakelite-treated felt, etc.).

72. Reducing the Vibrations Caused by Mechanical Sources

The main sources of low-frequency vibrations and mechanical noise in electrical machinery are the centrifugal forces produced by the static and dynamic unbalance of the rotors. Normally the rotors of small and medium power electrical machines are subjected only to static balancing.

Static balancing, however, cannot be done with a great degree of accuracy (about 5 microns of residual displacement of the rotor's center of gravity) (57). Moreover, static balancing cannot eliminate the

disturbing moments produced by dynamic unbalance. Therefore the electrical machines whose rotors have been subjected to static balancing alone can experience intensive vibrations both of the machine as a whole and of its separate parts.

The expected level of acceleration of sonic vibration at the supports of a vibration isolated machine can be determined by a formula from reference (63):

$$\beta_y = 20 \log F_v - 12 \text{ db} \tag{246}$$

where $F_v = \mathcal{E}_c \mu n^2 D_z$ - is the so-called vibration factor of the machine;

$\mathcal{E}_c = \dfrac{m_1 r_1 \pm m_2 r_2}{m}$ - the static unbalance of the rotor, cm;

r_1 and r_2 - the distances of the unbalanced masses from the rotational axis;

m_1 and m_2 - the unbalanced masses in the planes of balancing; the plus sign is used when the unbalanced masses m_1 and m_2 are on the same side of the axis; the minus sign is used when m_1 and m_2 are diametrically opposite;

n - is the rotor rpm;

μ - the ratio between the rotor mass m and the machine's mass M.

The quantity D_z is the coefficient of dynamicity of the machine and is determined from the following expression:

$$D_z = \beta_z + \frac{hA}{i_x^2} \beta_\varphi + \frac{\varepsilon_g l B}{\varepsilon_c i_y^2} \beta_\psi. \tag{247}$$

In this expression

$$\beta_z = \frac{\frac{\omega^2}{\omega_{0z}^2}}{\left(1-\frac{\omega^2}{\omega_{0z}^2}\right)^2+\gamma^2}; \tag{248}$$

$$\beta_\varphi = \frac{\omega^2}{\omega_{\text{Bx}}^2}\sqrt{\frac{\left[1-\left(1-\frac{h_0}{h}\right)\frac{\omega^2}{\omega_{0y}^2}\right]^2+\gamma^2}{\left(1-\frac{\omega^2}{\lambda_1^2}\right)^2\left(1-\frac{\omega^2}{\lambda_2^2}\right)^2+\xi_1^2}}; \tag{249}$$

$$\beta_\psi = \frac{\omega^2}{\omega_{\text{By}}^2}\sqrt{\frac{\left(1-\frac{\omega^2}{\omega_{0x}^2}\right)^2+\gamma^2}{\left(1-\frac{\omega^2}{\lambda_3^2}\right)^2\left(1-\frac{\omega^2}{\lambda_4^2}\right)^2+\xi_2^2}}; \tag{250}$$

$$\xi_1 = \gamma\left[2-\left(1+\frac{h_0^2}{l_x^2}\right)\frac{\omega^2}{\omega_{\text{Bx}}^2}-\frac{\omega^2}{\omega_{0y}^2}\right]; \tag{251}$$

$$\xi_2 = \gamma\left[2-\left(1+\frac{h_0^2}{l_y^2}\right)\frac{\omega^2}{\omega_{\text{By}}^2}-\frac{\omega^2}{\omega_{0x}^2}\right]. \tag{252}$$

In the latter expressions

ω - is the angular speed of the rotor;

γ - the loss coefficient of the material of the isolation mount;

h - the distance from the rotational axis to the plane of installation of the mounts;

h_0 - the distance from the machine's center of gravity to the plane of installation of the mounts;

l - the distance between the planes of balancing;

A and B - the distances between the center of stiffness O and the points of attachment of the mounts in the lateral and longitudinal directions;

i_x and i_y - the radii of inertia of the machine in relation to the x and y axes;

ω_{Bx} and ω_{By} - the circular frequencies of free rotational vibrations of the machine on the mounts in relation to the x and y axes;

ω_{Oz}, ω_{Oy}, ω_{Ox} - the circular frequencies of free vibrations of the machine in the directions of the z, y and x axes;

λ_1 and λ_2 - the circular frequencies of free double-coupled vibrations of the machine on the vibration isolation mounts in the zoy plane.

The λ_1 and λ_2 frequency values can be found from the following formula:

$$\lambda_{1,2}^2 = \frac{\omega_{0y}^2}{2}\left[1+\frac{\omega_{Bx}^2}{\omega_{0y}^2}+\frac{h_0^2}{i_x^2} \pm \sqrt{\left(1+\frac{\omega_{Bx}^2}{\omega_{0y}^2}+\frac{h_0^2}{i_x^2}\right)^2 - 4\frac{\omega_{Bx}^2}{\omega_{0y}^2}}\right]. \quad (253)$$

The λ_3 and λ_4 frequency values can be determined from that expression by changing ω_{Bx}, ω_{Oy} and i_y to ω_{By}, ω_{Ox} and i_x, respectively.

The quantity ε_d is called the specific dynamic imbalance in each balancing plane. This quantity is determined by the following formula

$$\varepsilon_d = \frac{m_1 r_1 a \pm m_2 r_2 b}{m \ell}$$

where the plus sign is used when the inertia forces of mass m_1 and m_2 coincide in direction, and the minus sign when the inertia forces of the masses are directed in opposite directions.

Formula (246) describes the vibration characteristics of a machine under the action of its own unbalanced forces. It can be used not only to calculate the sonic vibration level of machines on vibration isolation mounts, but also to determine the vibration of machines installed without such mounts on ship's foundations, which almost always have a finite

stiffness. In the latter case the calculation requires also a knowledge of the attached masses and the coefficients of stiffness of the foundation.

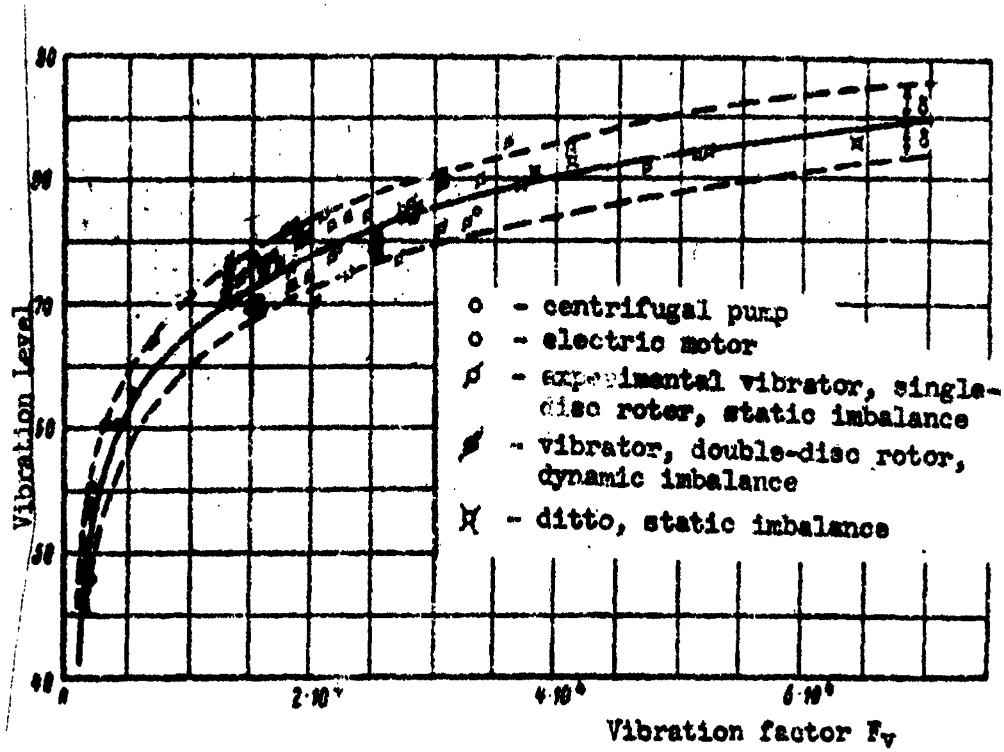


Figure 200. Relationship between the vibration level at the supports of the machine and the vibration factor F_v. Vibration levels are in decibels re 3×10^{-2} cm/sec^{-2}. $\delta = \pm 3$ decibels -- margin of error of vibration-measuring apparatus

Figure 200 shows the acceleration level of the vibrations at the supports of certain experimental machines and installations as a function of the vibration factor F_v. As may be seen from this graph, the deviation of the experimental points from the theoretical curve both in the case of static and dynamic imbalance does not exceed $\pm$ 3 decibels, that is, it is within the limits of error of the measuring instruments. An acceleration of 3×10^{-2} cm/sec^2 is taken as the zero vibration level.

Example 59. Determine the vibrational acceleration level at the supports of a machine (Figure 182) from the following data:

Q_{H1} = 5 grams; r_1 = 10 cm; Q_{H2} = 3 grams; r_2 = 12 cm; a = 15 cm; b = 20 cm; γ = 0.1; $m = \frac{30}{981}$ kg sec^2 cm^{-1}; $M = \frac{120}{981}$ kg sec^2 cm^{-1}; i_y = 15 cm; A = 20 cm; B = 25 cm; c_z = 4,000 kg/cm.

Solution. Inasmuch as in a machine of this type, $h = h_o = 0$, then according to formula (247), the coefficient of dynamicity will be

$$D_z = \beta_z + \frac{\varepsilon_R l B}{\varepsilon_c i_y^2}\beta_\psi .$$

In this case the vibrations of the machine in the direction of all the coordinates will be independent (uncoupled).

Let us determine the circular free vibration frequencies of the machine, which are required for purposes of the calculation.

The free vibration frequency of the machine in the direction of the z axis:

$$\omega_{0z} = \sqrt{\frac{c_z}{M}} = \sqrt{\frac{4000\cdot 981}{120}} = 181 \ \text{sec}^{-1}.$$

The frequency of the rotational vibration of the machine around axis y:

$$\omega_\psi = \sqrt{\frac{c_z B^2}{M i_y^2}} = \omega_{0z}\frac{B}{i_y} = 181\cdot\frac{25}{15} = 302 \ \text{sec}^{-1}.$$

The dynamic factor:

$$\beta_z = \frac{\frac{\omega^2}{\omega_{0z}^2}}{\sqrt{\left(1-\frac{\omega^2}{\omega_{0z}^2}\right)^2+\gamma^2}} = \frac{\frac{314^2}{181^2}}{\sqrt{\left(1-\frac{314^2}{181^2}\right)^2+0,1^2}} = 1,5.$$

In the same way we find the factor β_ψ (for the case of uncoupled rotational vibrations of the machine):

$$\beta_\psi = \frac{\frac{\omega^2}{\omega_{sy}^2}}{\sqrt{\left(1-\frac{\omega^2}{\omega_{sy}^2}\right)^2+\gamma^2}} = \frac{\frac{314^2}{181^2}}{\sqrt{\left(1-\frac{314^2}{302^2}\right)^2+0{,}1^2}} = 8{,}45.$$

The static rotor imbalance:

$$\varepsilon_c = \frac{m_1 r_1}{m} = \frac{5\cdot 10 - 3\cdot 12}{30\cdot 1000} = 4{,}67\cdot 10^{-4} \text{ см.}$$

The specific dynamic imbalance of the rotor:

$$\varepsilon_d = \frac{m_1 r_1 a + m_2 r_2 b}{ml} = \frac{(5\cdot 10\cdot 15 + 3\cdot 12\cdot 20)\cdot 981}{30\cdot 981\cdot 1000\cdot 35} = 14{,}1\cdot 10^{-4} \text{ см.}$$

The coefficient of dynamicity of the machine:

$$D_z = \beta_z + \frac{\varepsilon_d l B}{\varepsilon_c i_y^2}\beta_\psi = 1{,}5 + \frac{14{,}1\cdot 10^{-4}\cdot 35\cdot 25}{4{,}67\cdot 10^{-4}\cdot 18^2}\cdot 8{,}45 = 160.$$

The mass ratio:

$$\mu = \frac{m}{M} = \frac{30\cdot 981}{120\cdot 981} = 0{,}25.$$

The vibration factor:

$$F_v = \varepsilon_c \mu n^2 D_z = 4{,}67\cdot 10^{-4}\cdot 0{,}25\cdot 3000^2\cdot 160 = 1{,}680{,}000$$

The vibration level at the machine's supports:

$$\beta_y = 20 \log F_v - 12 = 20 \log 1{,}680{,}000 - 12 = 92.5 \text{ db}$$

The noise produced by an unbalanced rotor usually has a low frequency and does not greatly affect the human ear. But as was already pointed out, the disturbing forces produced by the imbalance have higher harmonic components which may fall in nicely with the natural frequency of one of the parts. This can sharply increase the machine's noise.

Moreover, a low frequency vibration can bring about a rattling of any loose metal parts or a very unpleasant modulation of the noise of other sources.

A careful dynamic balancing of the rotors on their own bearings on precision electronic balancing machines, the elimination of resonance regimes from the machine's operating range, an increase in the stator mass, and an increase in the distance between the machine supports makes possible a considerable reduction in the level of sonic vibrations in machines due to the imbalance of their rotating parts.

Roller bearings are another source of intensive mechanical vibration and noises in electrical machines.

Differences in the walls of the inside rings of bearings produce an unbalanced centrifugal force and a dynamic moment. The vibration of the machine in this case is similar to that occasioned by the presence of static and dynamic imbalances in the rotor.

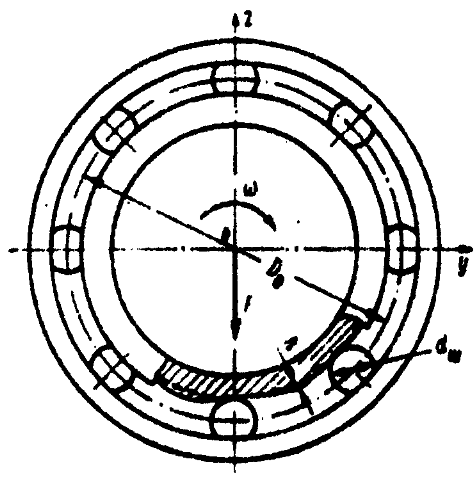


Figure 201. Rough spots "ripples" on the internal raceway of a roller bearing, which are responsible for bearing vibration and noise.

The noise and vibration of a bearing are particularly effected by the "ripples" (Figure 201) on the roller paths. Even insignificant "ripples" (0.5 micron high) can produce very intensive structure-borne and airborne noise. The fundamental vibration frequency depends on the relationship between the number of waves z_w and the number of balls z_b (62):

$$f_w = \frac{f}{2} \left(1 \pm \frac{d_b}{D_o}\right) \frac{z_b z_w}{q} \qquad (254)$$

where d_b - is the ball diameter

D_o - the diameter of the circle passing through the centers of the balls;

q - the greatest common denominator of z_b and z_w.

In this expression the minus sign is used for the external ring and the plus sign for the internal one. The frequency of these vibrations is mostly in the range of 500-5,000 cps, that is in the range of the natural frequencies of the machine's parts and in the region of the greatest sensitivity of the human ear.

The radial clearance in the bearing results in an uneven distribution of the loading on the balls and therefore to a varying displacement of the rotor at various positions of the balls in relation to the load. The result is a vibration of the rotor's center of gravity which also leads to the vibration of the machine. The vibration frequency produced by the i-th harmonic of the disturbing force is

$$f_z = \frac{f}{2} \left(1 - \frac{d_b}{D_o}\right) z_b \, i \qquad (255)$$

The extent of rotor vibration depends on the type of bearing and the relation between the loading and clearance in the bearing (64, 152). Investigations have shown that there are two optimal clearances in a bearing which can reduce the rotor vibration to zero, and that means a minimum of vibration and noise.

Another reason for the intensive vibration of roller bearings is an oval shape of the balls. The vibration frequency produced by ovality of the balls is

$$f_o = f \frac{D_o}{d_b} \left(1 - \frac{d_b^2}{D_o^2}\right) z_b \qquad (256)$$

Example 60. A rotor rotates on ball bearings type No. 207 at a speed of $n = 3,000$ rpm. The internal ball race has 30 "ripples". Determine the fundamental vibration frequencies produced by the wavy path of the internal ball race, by the bearing clearance, etc. Description of bearing: $z_b = 9$, $D_o = 57.5$ mm, $d_b = 11$ mm.

Solution. The vibration frequency produced by differences in the wall of the ring,

$$f = \frac{n}{60} = \frac{3000}{60} = 50 \text{ cps.}$$

The vibration frequency produced by the ripples (formula 254),

$$f_w = \frac{f}{2}\left(1 - \frac{d_b}{D_o}\right)\frac{z_w z_b}{q} = \frac{50}{2}\left(1 - \frac{11}{57.5}\right)\frac{30 \cdot 9}{3} = 2{,}680 \text{ cps.}$$

The vibration frequency produced by the clearance in the bearing (formula 255)

$$f_z = \frac{f}{2}\left(1 - \frac{d_b}{D_o}\right) z_b = \frac{50}{2}\left(1 - \frac{11}{57.5}\right) 9 = 182 \text{ cps.}$$

The vibration frequency produced by the oval shape of the balls (formula 256)

$$f_o = f\,\frac{D_o}{d_b}\left(1 - \frac{d_b^2}{D_o^2}\right) z_b = 50\,\frac{57.5}{11}\left(1 - \frac{11^2}{57.5^2}\right) \cdot 9 = 2{,}260 \text{ cps.}$$

As seen in the above example, the vibration frequencies occurring in roller bearings have a very wide range. It should also be borne in mind that each of the above-mentioned disturbing forces has a number of higher harmonic components and that vibrations of varying frequencies can occur also as a result of differing ball sizes, poorly machined parts and other causes.

To reduce the noise of roller bearings, all the above-mentioned inaccuracies should be avoided at all costs. Major attention should be concentrated on reducing the surface ripples of the internal and external ball races and the reduction of the non-uniformity and ovality of the rolling parts. Noise-tested and precision made bearings that tend to reduce noise and vibration should be used in marine ships and other types of transport vehicles. The use of quiet bearings in electric motors, as we have seen, makes it possible to reduce their noise level by 4-5 decibels (Figure 196). The substitution of Class B (cyrillic) bearings for ordinary bearings results in a reduction of sonic vibrations (in acceleration) by 20-25 decibels (Figure 202).

Theoretically, small diameter bearings with a large number of rolling bodies are less noisy than large diameter bearings with fewer rolling bodies. "Needle-type" [igol'chatyye] bearings are among the quietest of roller bearings.

Large clearances in the seats of the bearing retainers produce "retainer noise". This can be reduced by making the separators (bearing retainers) from materials with a high damping characteristic (polyamidic resins, textolite). The retainer must have a limited "floating clearance" and minimal clearances in the ball seats.

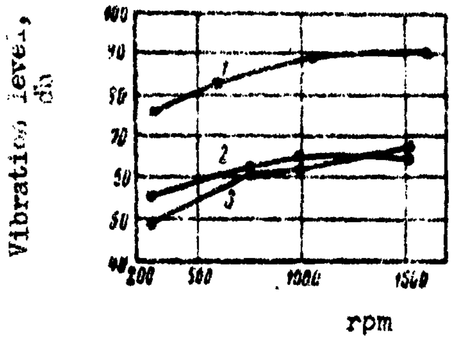


Figure 202. The effect of the type of bearing on the level of vibrational accelerations of a ship's electric converter.

1 - class H [cyrillic] ballbearings;
2 - class B [cyrillic] ballbearings;
3 - plain bearings.

The noise produced by roller bearings is to a large extent determined by the carefulness of their installation in the machine, particularly by the way the rings are placed and fitted onto the shaft and into the body of the machine, and the cleanliness of the journal. It is recommended that the bearings be fitted tightly onto the shaft journals and into the seats of the bearing's housing (75). The seats must be well machined with a minimum deviation from a true cylindrical form. The bearing should be installed without its being canted or pinched.

The noise of the bearings is also affected by the care given them in the course of their operation. An unlubricated (dry) bearing will produce a great deal of noise because of the intensification of vibration of its parts due to dry friction. The noise of a heavily lubricated bearing in an engine is about 3 decibels lower than that of a dry bearing. A dirty bearing will increase the vibration and noise of the machine, especially in the high-frequency range.

Theoretical and experimental researches have shown that the noise of serially produced roller bearings cannot be reduced below a certain limit (65-70 decibels). A further reduction in the noise level would require a much higher machining precision of the parts, which is technically difficult and economically impractical. In view of that circumstance, noise control should be carried out on the machine itself. To reduce the airborne noise radiated directly by the bearings, it is recommended that they be closed with tight-fitting covers to prevent the propagation of the noise into the surrounding medium. If a high noise level is caused by intensive resonance vibrations of the parts, the resonance should be tuned out by increasing the stiffness of the parts.

Good results can be achieved in the medium and high-frequency ranges by the introduction of vibration- and sound-isolating devices between roller bearings and the stator. Intermediate bushings made of elastic materials, such as rubber and plastics, should be used for that purpose. This would reduce the total level of sonic vibration and noise by 9-12 decibels. Bushings and gaskets made of paper, cardboard, babbit or lead, as recommended by some researchers, do not reduce noise and vibrations to any appreciable extent.

Plain bearings should be used for particularly quiet machines. This produces a considerable reduction in the noise of the high-frequency spectral components.

The bearing noise level tends to rise after lengthy operation due to the uneven wear of the bearing seats and the appearance of waves on the bearing surfaces. The proper maintenance of the bearings in the course of operation and the timely replacement of worn-out bearings (wherever possible) are therefore necessary preventive anti-noise measures to be carried out in present-day shipboard practice.

A third source of mechanical noise in electrical machines are the brushes and the commutator. As the rotor revolves, the commutator plates impact upon the brushes and produce a vibration of the brush holders and the brushes themselves. The frequency of the brush noise is determined by the following formula:

$$f_{br} = f z_k i \tag{257}$$

where z_k - is the number of commutator plates;

$i = 1, 2, 3...$ - the noise harmonics.

To reduce the brush noise, the commutator should be very well pressed together and its surface properly and cleanly finished; the deviation of the commutator surface from a true cylindrical form should be reduced to a minimum. Deeply grooved (eroded) commutators are - as a rule, noisier than those with shallow grooves or without them.

The brush holder and brushes should be fairly rigid with a minimum clearance between them (113). Well ground smooth brushes made of soft materials can reduce the noise by 8-10 decibels.

73. Reducing the Cooling Fan Noise of Electrical Machines

A direct cause of intensive ventilation noise in electrical machines is the periodic vortical motion of the air flowing through it. The amount of vortex-formation is determined by the design of the fan and

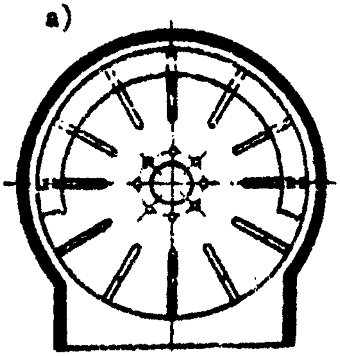
the speed of the air flow. Inasmuch as the speed of the air flow in a machine seldom exceeds 8-10 meters per second, major attention should be focused on the design arrangement of the fan.

The investigation of electrically driven pumps, carried out by S. Ya. Novozhilov, shows that the noise produced by the fan of a motor is 6-10 decibels higher than that coming from other sources in the machine. A radial gap in the fan which is too small produces a strong disturbance in the air flow resulting in an unpleasant "siren"-type noise. As the gap is widened, the "siren" noise is reduced. An optimal radial gap is 10-15% of the ventilator diameter. A further increase in the gap would markedly reduce the fan output without any appreciable effect on the noise.

An increase in the fan diameter (at a given rpm) causes an increase in the peripheral velocity, resulting in a higher noise level. When designing a fan, one should select a minimum diameter.

As an example of an unsuccessful design solution, there is presente in Figure 203-a the original arrangement of the cooling fan blading in a PNV-68 electric motor, which is used to drive certain types of shipboard pumps. Two thirds of the peripheral section of the blading do not participate in the work as there is no outlet for the air in this area. This reduces the output to a third of the normal level. At the same time, too small a clearance between the blading and the housing of the fan produces a great deal of noise.

a)



b)

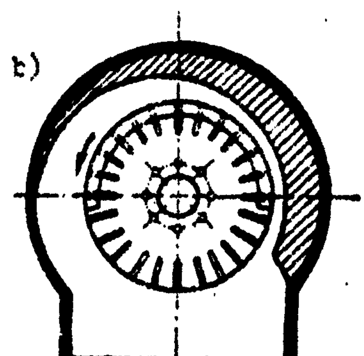


Figure 203. Original (a) and improved (b) cooling fan design of a PNV-68 electric motor

The cooling device of this electric motor has been improved with a view to reducing its noise. The fan originally having an external diameter of 296 mm was replaced by a 195 mm-diameter fan with twice as many blades. An intermediate body of variable thickness was installed near the cooling fan blading forming a spiral duct (Figure 203-b). This measure reduced the motor's noise by six decibels. A similar improvement of the fan in an MAF 83-74/2 electric motor reduced its noise level by 9 decibels.

The structure of the fan itself must be sufficiently rigid. In some cases it is found worthwhile to damp the blading by surface friction (in hollow riveted blades) or by vibration-absorbing coatings.

Fan noise can be drastically reduced by radically changing the principle of the fan's operation (by the use of tangential fans; see the next chapter).

Aerodynamic noise originates not only in the machine's fan but also as the air flows around all the rotating parts of the machine. To reduce fan noise, it is recommended to streamline the leading edges of parts of the rotor and its entire external surface over its entire length.

The silencing of an electrical machine calls for the simultaneous elimination of noises coming from different sources. An example of such overall silencing . the 2.5 kw. electric motor of the CR series, demonstrated by the Siemens Company at the "Less Noise" exhibition of the Third International Congress on Acoustics. Plain bearings and spray lubrication (by means of a ring immersed in an oil bath) were used in that motor. Cooling openings were abolished, the heat being transferred into the air through 30 mm-high fins installed on the motor body. The rotor shaft was installed in a mount on rubber rings; moreover the motor also had very soft vibration isolation mounts (free vibration frequency 1-2 cps); measures had been taken to reduce the magnetic noise.

All the above-listed measures combined to reduce the noise of the motor to less than 30 decibels. The only way one could tell whether the motor was working, without absolute quietness in the building, was by the little glitters of the shaft (about 1,420 rpm). The use of such motors for ship's service purposes (in ventilators, air conditioners and various pumps) would permit a substantial reduction in the noise of such systems on shipboard.

It is much more difficult, of course, to achieve such a sharp reduction in the noise level of electric machines of high power. But even in this case it would be possible to reduce the noise 10-15 decibels below the level of existing machines without a substantial increase in cost by using the whole gamut of measures designed to reduce noise at the source.

CHAPTER 19

CONTROLLING THE NOISE OF SHIPBOARD VENTILATORS,

PUMPS AND SHIP'S SERVICE SYSTEMS

74. General Characteristics of Ventilator and Pump Noises

The noise of ventilators varies from 70-80 to 100-120 decibels, depending on their capacity and design. The ventilation noise spectra lie within a wide range of frequencies (from 25 to 6,000 cycles per second).

One of the chief causes of ventilation noise is vortex formation in the air channels of the ventilator. The sound power of the vortical noise emitted by the blade wheel of a ventilator is defined by the following expression:

$$W = k \frac{\rho}{c^3} U^6 D^2 \qquad (258)$$

where k - is the coefficient that takes into account the geometrical form of an object in a flow, the direction of flow and the Reynolds and Mach numbers;

U - peripheral velocity of the blade wheel;

D - diameter of the blade wheel;

ρ and c - density of the medium and the speed of sound in it.

Formula (258) shows that the chief factors affecting vortical noise is the peripheral velocity of the blade wheel and its size. Vortex formation in a working medium is intimately connected also with the frontal resistance* of a body. The k factor is reduced when the frontal resistance

* "frontal resistance" is the Russian expression for total drag - [Translator]

of the body is reduced. Consequently, in principle, the higher the efficiency of the ventilator, the lower its vortical noise level.

The vortical noise frequency is determined primarily by the frequency of vortex break-away, which, in turn, depends on the transverse dimensions of the body and the speed of the air flow. This frequency can be found from the following expression:

$$f_v = Sh \frac{U}{d \sin \alpha} i \qquad (259)$$

where $\underline{Sh}$ - represents the Strouhal number;

d - a geometrical dimension of the body;

α - the angle of its position (Figure 187);

i - 1, 2, 3, ...

The Strouhal number Sh is determined experimentally; in the case of a plate and a cylinder, $\underline{Sh} = 0.18 - 0.2$.

Of major importance in a vortical noise spectrum are the first harmonic components. However, since the speed and direction of the flow in the ventilator constantly change along the blades, the individual blade elements emit a noise of different frequency. This explains the continuous noise spectrum in a wide band of frequencies.

Another source of aerodynamic noise in ventilators is the nonuniformity of the flow of the working medium. Inasmuch as the nature of this noise is similar to that of vortical noise, the sound power produced by nonuniform flow can also be determined by formula (258), the only difference being a different k value.

The frequency of the noise produced by nonuniform air flow is determined by formula

$$f_n = f z_b i \qquad (260)$$

where z_b is the number of blades;

i -- 1, 2, 3, ...

The first (fundamental) harmonic is the most important one. As mentioned earlier, the noise produced by a nonuniform air flow is called "siren" noise, because of its unpleasantness.

The mechanical noise of ventilators is caused primarily by rotor imbalance in the driving motor (with its cooling fan) and the vibration of the antifriction bearings. Both of these factors can produce intensive resonance vibrations of the exterior walls and driving parts. With an increase in the peripheral speed of the ventilator, the aerodynamic noise becomes more intensive than mechanical noise and is almost always predominant. Inasmuch as the peripheral speed of shipboard centrifugal ventilators is 10-15 meters per second, their mechanical noise may be disregarded.

The noise level of marine ventilators can be defined by the following formula (15):

$$\beta = 14 \log \frac{QH}{\eta} + 6 \text{ db} \qquad (261)$$

where Q, H and η - represent the ventilator capacity, pressure and efficiency.

The values of β obtained from formula (261) correspond to measurements taken at a distance of 0.5 meter from the ventilator.

Example 61. Determine the noise level of a ship's centrifugal ventilator having the following characteristics: $Q = 800$ m^3/hour, $H = 60$ mm of water and $\eta = 0.62$.

Solution. Using formula (261), we find:

$$\beta = 14 \log \frac{QH}{\eta} + 6 = 14 \log \frac{800 \cdot 60}{0.62} + 6 = 75 \text{ db.}$$

Figure 204 shows the graph of formula (261). The points correspond to experimental data for ships' ventilators with various values of the parameter $\frac{QH}{\eta}$.

Similar causes of noise are in principle inherent also in centrifugal and propeller-type pumps. But the cavitation of the working fluid in pumps is, as a rule, a greater source of sonic vibration and noise.

Figure 205 shows a diagram of the noise-forming sources in a centrifugal pump. The intense vortex formation during the flow of a fluid in a centrifugal pump is due to the break-away of the flow from the blade surface (zone a), the formation of a surface of discontinuity where the flow leaves the blades (zone b), and the existence of circulation at the entrance to the impeller and at pressure relieving openings (zone c). In this vortex formation process, the pressure in some vortices may drop below the saturation pressure resulting in the liberation of the dissolved air from the water (air cavitation).

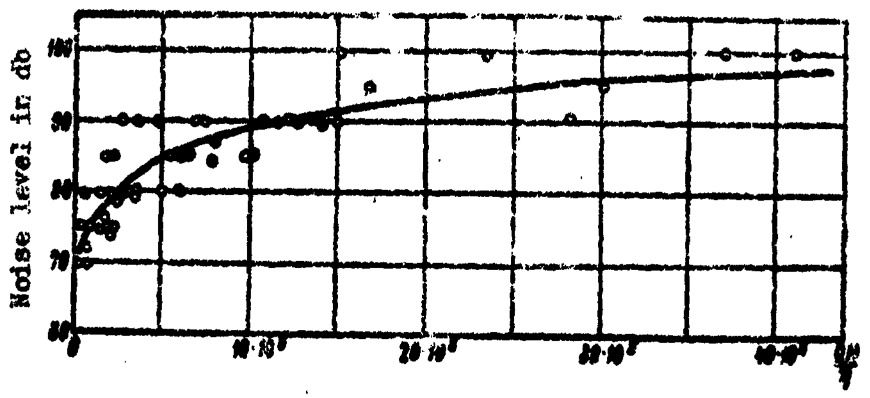


Figure 204. The relationship between total noise level and capacity for shipboard ventilators. The curve is based on formula (261), and can be used for calculations.

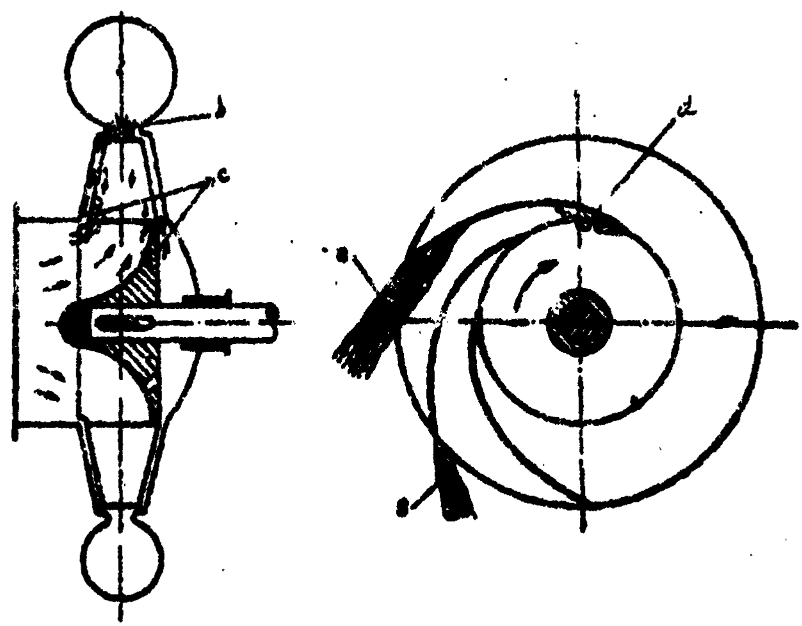


Figure 205. The vortex-formation and cavitation zones in the wheel of a centrifugal pump.

If the operation of the pump occurs with a low suction pressure, the pressure on the suction side (at the inlet edge) of the blades may drop to the vapor-formation level, the resultant boiling of the liquid producing "steam" (vapor) cavitation (zone d, Figure 205). As was mentioned above, both types of cavitation are conducive to intensive noise formation, mostly at frequencies over 1,000 cps. The lower the suction pressure, the higher the cavitation noise level of the pump. Figure 206 shows the relationship of the discharge pressure H_n and the level of sonic vibrations of a centrifugal pump to the suction pressure and air content of the water, as obtained by G. A. Khoroshev. As the suction pressure p is reduced, cavitation develops further and becomes an increasingly intensive source of vibration.

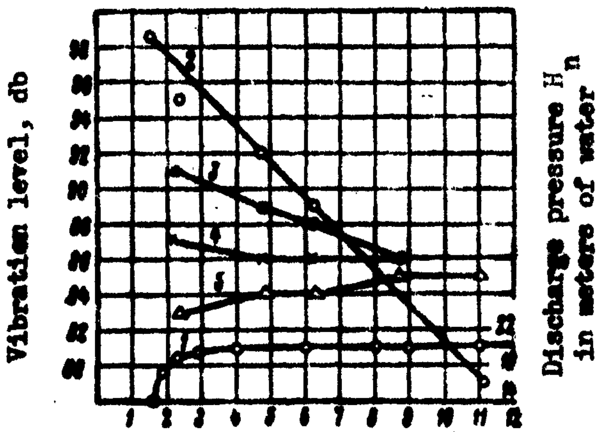


Figure 206. The relationship of the vibration of a centrifugal pump to the suction pressure and the relative air content of the water.

Curve 1 - Discharge pressure H_n vs suction pressure p.

Curves 2 to 5 - Vibration level β vs suction pressure p and the air content q of the water. Curve 2, q = 0; Curve 3, q = 0.0073; Curve 4, q = 0.055; Curve 5, q = 0.186.

75. Reducing the Noise of Ventilators and Pumps by Changing Their Design Parameters

The most effective method of reducing the noise of ordinary ventilators is to reduce or limit the peripheral speed of the blades. A tip speed not exceeding 15-20 meters per second is suggested for centrifugal ventilators.

Another design factor affecting the noise of a ventilator is the size of the blade wheel. The change in ventilator noise level produced by altering the blade diameter from D_1 to D_2 may be found in the following expression:

$$\Delta \beta = 20 \log \frac{D_2}{D_1} \text{ db} \tag{262}$$

The ventilator noise is very largely affected by obstacles at either the inlet or outlet side of the wheel, which produce a nonuniform air flow. To reduce the noise produced by such an air flow in axial ventilators, the bearing supports should be streamlined and a commutator with a gradually changing cross section should be installed at the inlet.

The noise of centrifugal ventilators produced by a nonuniform air flow is frequently caused by too large a ventilator tongue. The determining criterion in this matter is the distance between the blade wheel and the tongue. Located too close to the wheel, the tongue can raise the noise level by 12 decibels. The distance between the tongue and the wheel should therefore be at least 10-15% of the blade diameter. The noise depends on the shape of the ventilator tongue. A tongue with a skewed section over its whole length is the most effective noise reducer.

The noise of centrifugal ventilators is determined by the number of blades. In some cases an increase in the number of blades can reduce the noise by 3-5 decibels.

The noise of a ventilator can be substantially reduced (by 7-8 decibels) by improving its air-flow passages. Figure 207 shows a diagram of a "straight-through" ventilator, and Figure 208 the results of tests on it.

One of the characteristics of a straight-through ventilator, according to Figure 207, is that its driving motor is located inside the ventilation channel itself. The noise of a straight-through ventilator, as a unit, was found to be 7 decibels lower than that of an ordinary ventilator.

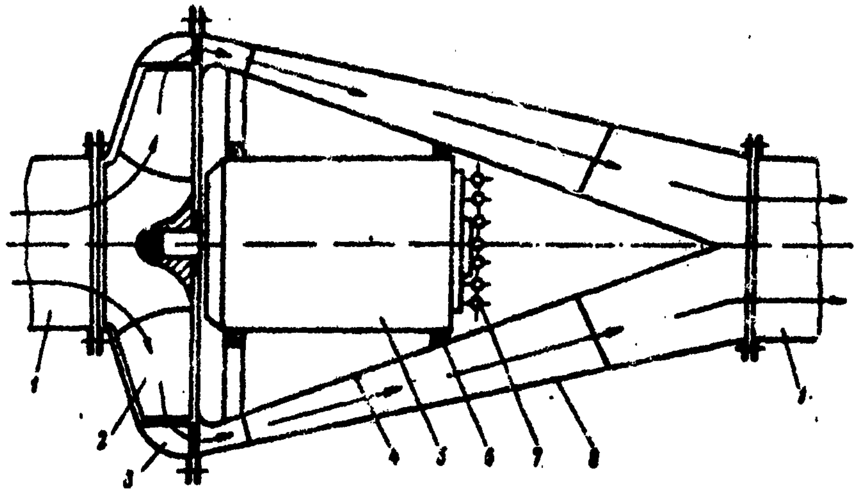


Figure 207. A "straight-through" electrically driven ventilator.

1 - air inlet duct; 2 - ventilator blading; 3 - guide vane; 4 - motor housing; 5 - electric motor; 6 - rubber gaskets; 7 - motor-cooling vents; 8 - ventilator casing.

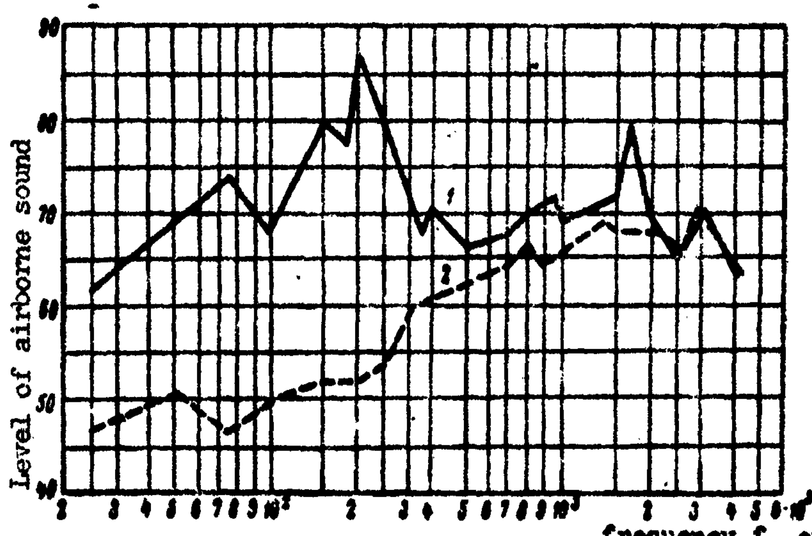


Figure 208. Noise spectra of a normal (1) and a "straight through" (2) ventilator with an output Q = 2,000 cubic meters per hour.

Axial ventilators are faster, and the noise they produce at the same output capacity is characterized by higher frequencies and is therefore more unpleasant than the noise of centrifugal ventilators.

When the Reynolds numbers are low ($Re < 80{,}000$), the lift coefficient is reduced and the total drag coefficient is increased. The result is a lower efficiency. Working within the range of low Reynolds numbers are small ventilators with low peripheral speeds as well as ventilators designed for light and rarified gases. The efficiency of such ventilators is usually very low (1-10%). The Laing Company (146) has recently developed a special type of ventilator to work at low Reynolds numbers. These have come to be known as "tangential" ventilators

The "delivery convergence" principle is used in these ventilators to increase their efficiency. Under the new principle, unlike ordinary ventilators, different quantities of air pass through the different channels of a circular set of blades (profiles), and most of the gas flow is concentrated within a very narrow arc of the channels. This leads to a considerable increase in local velocities and, consequently, higher local Reynolds numbers and greater ventilator efficiency. The distinctive characteristic of the new ventilators is that the air flows transversely toward the axis, and twice flows through the blading, turning itself by almost 180° (Figure 209).

Operating at very low Reynolds numbers, such ventilators can develop a considerably greater efficiency than the best of ordinary machines. The exhaust velocity of a tangential ventilator is considerably higher than the peripheral velocity. Thus at the same exhaust velocities, the peripheral speed of tangential ventilator is only 7-9% of the peripheral speed of ordinary ventilators.

The result of these favorable qualities of tangential ventilators is that their noise if considerably lower (by 25-30 decibels) than that of ordinary ventilators. Figure 210 shows the loudness level as a function of exhaust velocity for axial and tangential ventilators. Moreover, tangential ventilators are very small in size. The ventilators produced by the Laing Company are now being used for pressurizing and exhausting systems, for cooling and air-conditioning systems, for drawing off gas leakage, for boiler-combustion chambers, etc. The wide use of these ventilators on shipboard is to be expected.

The measures for reducing the noise of centrifugal and axial pumps are in large part similar to the above-discussed methods for controlling the noise of ventilators.

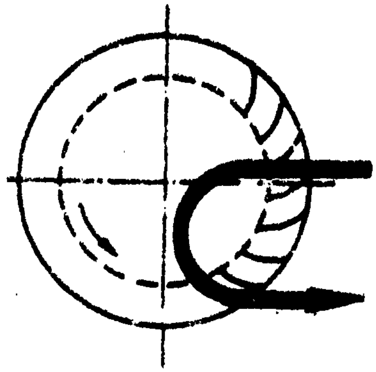


Figure 209. The direction of gas flow in tangential ventilators.

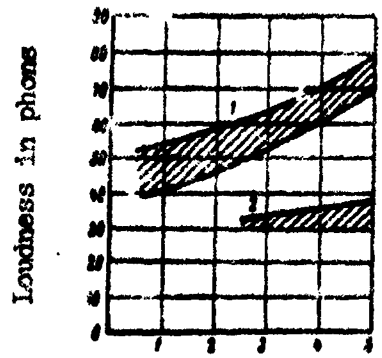


Figure 210. The relationship between the loudness level of ventilators and their exhaust velocity.

1 - axial ventilator;
2 - tangential ventilator.

For the noiseless operation of a centrifugal pump, it is necessary that it work at the point of maximum efficiency. At this point, the pump's operation is characterised by a minimum of hydraulic impacts, because the exit angle of the flow from the impeller blades corresponds to the designed angle. The farther the pump's operation from its maximum efficiency, the greater the vortex formation and the hydraulic impacts, and the higher the noise level.

A decrease of the cavitation noise in a pump calls for the following: a reduction in the radial velocity at the inlet to the impeller by enlarging the inlet; the use of thin double curvature blades having a small relative thickness, and also curving the blade profile at the inlet (sickle-shaped blades).

An excessively large diameter of the pump inlet, intended to reduce the flow velocity, may intensify the noise in the pump rather than reduce it. To develop into a liminat flow, the flow at the inlet of the pump should be as close as possible to the centerline of the shaft (126). A considerable increase in the suction diameter may be recommended only for condensate pumps and pumps working with hot liquids, where there is a possibility of boiling of the liquid.

Cavitation noise in pumps produced by vapor cavitation can be considerably lessened by admitting a small quantity of air near the inlet. (See Figure 206). The air bubbles can damp the vibration of the

cavitational bubbles, and also reduce the intensity of the sound impulses. Moreover, the air bubbles also disperse (to a certain extent) the high frequency sound waves. This contributes to a reduction in the vibration and noise of the pump unit.

Care should be taken to prevent cavitation during the pump's operation, and efforts should be made to have the pump work with a support [Russian: podpor].

76. The Application of Vibration Isolation and Vibration Absorption Treatments in Ventilators and Pumps

Some reduction in the noise of ventilators and pumps can be achieved by the use of vibration- and sound-isolation treatments.

Since it is made of thin sheet metal, a ventilator casing can produce intensive resonance vibrations under the action of disturbing forces generated by roller bearings and aerodynamic forces. This increases the noise of the ventilator.

The vibration and noise of the casing, caused by the motor drive, can be reduced by introducing an elastic connection between the electric motor and the shell. A vibration-isolating gasket of rubber or other elastic material can serve this purpose. This idea was incorporated in the design of a "straight-through" ventilator (Figure 207), and the vibration of its supports was thereby reduced 10 decibels below that of a conventional electric ventilator.

The resonance vibrations of the casing can be reduced by covering its surface with a damping coating. To achieve this, the production of ventilator casings from plastics is worthwhile.

Rubber bearings, which are effective as a vibration-isolating insert between the rotor and the pump casing, are now being used in hydraulic machinery. Such bearings reduce the transmission of disturbing forces to the casing and therefore decrease the sonic vibration of the machine.

Rubber or plastic insets have been successfully used in valve seats to reduce the valve impact in piston pumps. The valves themselves can also be made of such materials. They are not only less noisy in operation but, according to published data, have a longer service life.

The pump should be thoroughly vibration-isolated from the hull structure of the ship by the use of appropriate isolation mounts. Vibration isolating rubber sections, measuring 5 to 15 pipe diameters in

length, should be installed in the inlet and discharge lines of the pump. It is not recommended to place isolation mounts under each separate mechanism of a pump unit whose drive is independently installed and connected to it only by a coupling. The uneven load and vibration of the pump and its driving motor could put them out of alignment and raise the noise level. In this case a single-so-called "floating frame", installed on vibration isolation mounts, should be used for both the pump and its power source.

77. Fighting the Noise of Ships Plumbing Systems at the Source

The intensity of the noise produced by the housekeeping facilities on shipboard can reach some 70-80 decibels and, being sporadic in nature, it produces a very unpleasant effect upon the passengers and crew. This noise is particularly unpleasant at night when the general noise level in the staterooms and cockpits is relatively low. Such noise may originate from the water pipes, sewage system, heating system, bathrooms, showers, etc.

The usual sounds produced in water pipes are a kind of murmering noise, which is sometimes loud. This is caused by turbulent flow in the pipes or the swirling of the water in the faucets, sinks, washbasins and bathtubs. Air volumetric resonance also frequently occurs in the wash tubs, sinks and bathtubs which raises the noise intensity. The so-called "backlash" (water hammer) noises produced by closing the faucets or similar devices are unpleasant.

Vortex formation and cavitation in the pipelines and faucets are another powerful source of noise. The reason for these processes are sharp elbows and faulty faucet construction. The most widely used type of water pipe faucet (disc type) is usually quiet when the water outflow is very slow; but intensive vortex formation and noise begin when the water reaches a certain speed. This noise rises sharply when the open valve is in a certain position for which the frequency of vortex formation coincides with that of the natural vibration of the pipes in their hangers. The intensive resonance vibrations of the pipe produce a loud roar and rattling noise, especially if it is inadequately fastened.

Cavitation originates also from the discontinuous flow around the valve disk, which produces vapor and air bubbles. As they move, these bubbles collapse with a loud crack which is carried over the pipes for considerable distances.

To reduce the noise of various origins in the pipelines, the speed of the water flowing in them should be reduced to two meters per second. The water pressure in the systems used for housekeeping purposes should not exceed two atmospheres.

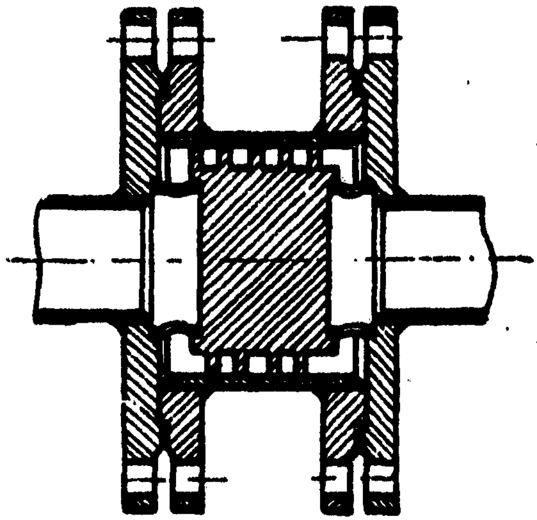


Figure 211. A noiseless helical throttle.

The marring noise produced by faucets and valves could be eliminated by abandoning the widely used valve with a flat disk. It should be replaced by a noiseless valve with a streamlined disk and seat. If the old disc-type faucet is still in use, the disc should be fastened tightly to the stem. But under no circumstances should fast-working plug-type faucets be used. The reduction of pressure in the faucet should be gradual, not sudden, and this helps reduce the noise.

Water and oil pipes should be equipped with throttling devices which have a gradual change in speed and pressure. The noiseless throttle developed for hydraulic installations in ships (99) operates on that principle. The gradual reduction in pressure in this case is achieved by having the oil flow along a helical channel (Figure 211). The speed of flow thus remains constant, and the drop in pressure is achieved by the lengthening of the throttling channel.

The total airborne noise level of a helical throttle does not exceed 70 decibels. For the same pressure drop and equal output, the noise of a helical throttle is 15-20 decibels lower than that of a diaphragm-type throttle, and 20-25 decibels lower than that of an ordinary throttle. Such throttles can be successfully used also for water piping.

The noise of washstands can be reduced by making the delivery pipes considerably smaller than the drain pipe. This prevents the over flow of water in the sink and eliminates the noise produced by a large amount of water as it runs down the drain; the drain pipe should leave the sink below the level of the stopper to prevent the possible noise formation by the air that might get into the drain pipe.

The noise produced by toilet [vaterklozet] facilities is caused by the rushing water down the chamber and the rising water under the float. To reduce the noise of the flushing water, the water tank should

be placed close to the top of the chamber and equipped with a push-button (rather than pedal) control. The noise of the water entering into the tank could be reduced by the use of a plastic diffuser between the pipe opening and the bottom of the tank. The valve device should be streamlined. A noiseless flushing device involving the use of the above-listed measures was demonstrated at the Third International Congress on Acoustics.

Much attention should be devoted to the arrangement of the pipes and the planning of sanitary and daily housekeeping service compartments. Vertical pipes should not be placed near the walls of passenger cabins. The installation of any kind of sanitary facilities near the bulkheads of adjoining staterooms should also be avoided. It is always desirable to have the various pipelines fastened to the ship's structure by elastic vibration-isolating supports (See also paragraphs 52 and 58).

CHAPTER 20

THE ACOUSTIC TESTING OF MACHINERY

78. Chambers and Test Stands for the Acoustic Testing of Machines, and the Methods of Testing

Theoretically, both reverberation chambers and sound damped chambers can be used at the machinery manufacturing plants for measuring the noise level of machinery and equipment. The former make it possible to determine the average value of the density of sound energy radiated by a machine, and the latter the spatial characteristics of the noise directionality and average noise level of a machine. The second type of chamber is usually preferable.

The chambers should be located as far away as possible from any source of noise and have well sound-isolated walls (2-3 courses of bricks thick). In the better types of chambers, the internal box (the actual chamber) is installed on vibration isolation mounts within a building with massive walls as shown in the diagram in Figures 212 or 102. The chamber is usually equipped with sound-isolated double doors.

The walls and ceilings and wherever possible the floor are covered with sound-absorbing materials. The latter may consist of wedges or cones of cotton wadding, glass felt, special plastics (fiberglass) and microporous rubber. The thickness of the sound absorber δ (in meters) is related to the lowest frequency f_m of the measurements envisaged in the chamber as follows:

$$\delta \approx \left(\frac{1}{4} \text{ to } \frac{1}{6}\right) \frac{c}{f_m} \approx \frac{70}{f_m} \qquad (263)$$

where c represents the velocity of sound in air.

Thus when f_m = 100 cycles per second, the sound absorber is three-quarters of a meter thick. This thickness should be taken into account when designing the chamber.

In sound damped chambers it is possible not only to measure the noise levels of machinery but also to trace the noise formation to the individual elements of the machine and check the effectiveness of work

on the noise reduction of machines. In the latter two cases the machine is put into operation by external power sources which are acoustically isolated from the machine under test (Figure 212). Finally, it is also possible to measure in these chambers the vibration isolation of isolating devices and the drop in the vibration level at vibration isolation mounts.

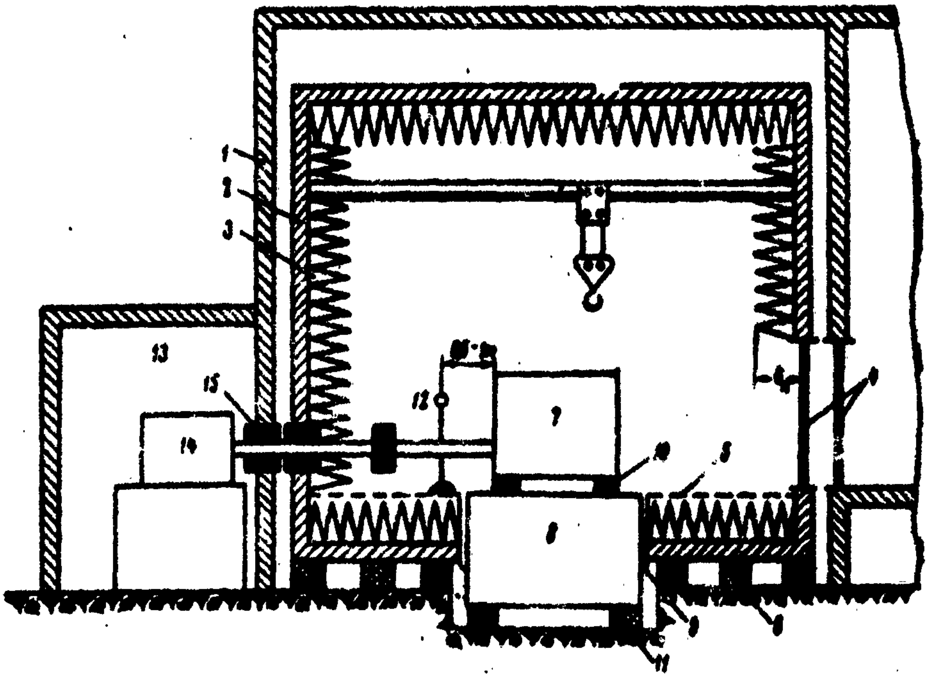


Figure 212. Sound damped chamber for measuring and investigating the noise of machines

1 - outer building; 2 - chamber; 3 - sound damper; 4 - double sound-isolated door; 5 - steel screen (chamber floor); 6 - vibration isolation mounts; 7 - machine under test; 8 - foundation; 9 - sound isolating packing; 10 - vibration isolation mounts of the machine; 11 - foundation vibration isolation mounts; 12 - measuring microphone; 13 - additional chamber for external drive; 14 - external power source; 15 - sound isolated lead-in of external drive shaft

The measurement and analysis of the noise produced by a machine are made under normal operating conditions under load. The measurements are made at 3-4 points at distances of 0.5-1 meter from the machine, and repeated 2-3 times. The results of the measurements are then averaged. When the difference between the noise levels measured at the various points does not exceed 4 decibels, the arithmetical mean of the results of measurements at all the points can be taken. If the scatter in the noise level readings at the various points is within the limits of 5-10 decibels, a power (square) summation should be used for determining the average level. The approximate value of the average sound level in this case would be

$$\beta_{cp} = \beta_{cp_a} + 2 \text{ db} \tag{264}$$

where β_{cp_a} -- is the average level obtained by arithmetical summation.

When the scatter in the noise levels at various points is greater than 10 decibels, the reason for the discrepancy should be investigated (interference effects, greater noise emitted by the machine in certain directions, etc.).

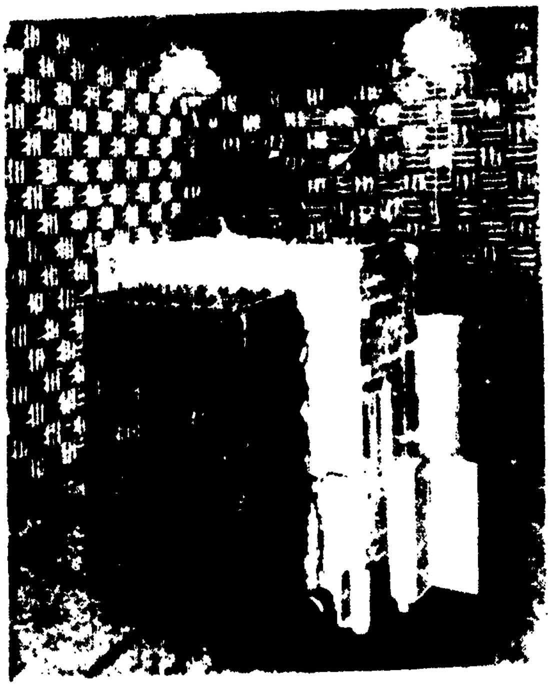


Figure 213. Testing a transformer in a sound damped chamber.

It is occasionally necessary to compare the results of noise measurements made in chambers with those made on shipboard. There is, as a rule, less sound absorption in ships' compartments than in sound-measuring chambers, and the noise level in sound-measuring chambers will therefore be less than that in the ship by the following amount:

$$\Delta = 10 \lg \frac{A_k}{A_c} = 10 \lg \frac{\alpha_k S_k}{\alpha_c S_c}, \tag{265}$$

where A_k, α_k and S_k represent respectively the values of the absorbtion, the average coefficient of absorption, and the average area covered by the sound-absorbing material in the chamber.

The similar quantities with a "c" subscript apply to the ship.

The acoustic interferences (unwanted noise) encountered when measuring the noise in ships are considerably greater than in special chambers. Such measurements are considered reliable only if the measured levels (the noise level of the machine plus interference) exceeds the level of the interference at corresponding frequencies by 10 decibels or more. In exceptional cases, when that requirement cannot be met, a correction chart is used (Figure 214).

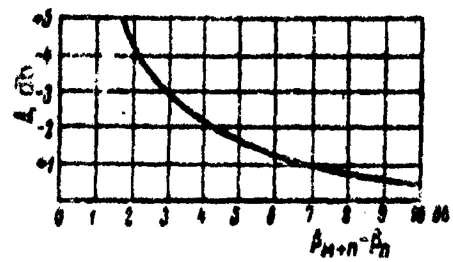


Figure 214. Correction factor for noise gage readings at high interference levels. The difference in the noise gage readings when a machine is operating and when it is not operating is plotted on the horizontal axis.

Example 62. The noise level of a machine being tested on a test stand in the manufacturing plant is 85 decibels. The interference level when the machine is not operating is 82.5 to 83 decibels. What is the true noise level of the machine?

Solution. The differences in the overall noise level (the noise of the machine plus interference):

$$\beta_{m+n} - \beta_n = 2 \text{ to } 2.5 \text{ db.}$$

According to the curve in Figure 214, the correction is $\Delta = -4$ decibels. The true noise level of the machine is: $\beta_m = 85 - 4 = 81$ decibels.

The measurements and analysis of noise in chambers can be successfully combined with the measurements and analysis of sonic vibration. This calls for a nonresonant foundation under the chamber, such as a massive concrete block, "decoupled" from the surrounding structure by vibration-isolating gaskets. The frequency of free vibrations of the foundation on such gaskets should be 3-4 times lower than the minimum vibration frequency to be measured on the machine. Only under that condition can the foundation be considered as separated from the surrounding structure during low frequency measurements.

Vibration levels can be measured with respect to the vibrational acceleration or vibrational velocity. As in the case of noise measurement, the mean figure of a number of vibration measurements at various

points of the machine or foundat..on is taken. One difficulty in comparing the measurement results in that no zero threshold has yet been established for vibration, and therefore the various measuring instruments have different zero levels (3×10^{-2} cm/sec^2, 3.5×10^{-2} cm/sec^2, 6.31×10^{-4} cm/sec^2, etc.). A vibrational acceleration of 3×10^{-2} cm/sec^2 (see paragraph 18) is herewith recommended as the zero level.

Let us assume that a vibration measuring instrument with a different zero threshold (which we shall designate as $\ddot{y}_{02}$) gives a vibration level of β_2. Then the vibration level β_1 with respect to the recommended threshold $\ddot{y}_{01} = 3 \times 10^{-2}$ cm/sec^2 will be

$$\beta_1 = \beta_2 \pm 20 \log \frac{\ddot{y}_{01}}{\ddot{y}_{02}} = \beta_2 \pm \Delta \text{ db} \qquad (266)$$

where the correction is

$$\Delta = 20 \log \ddot{y}_{02} + 31 \text{ db} \qquad (267)$$

The plus sign before the correction is used when $\ddot{y}_{02} < \ddot{y}_{01}$, and the minus sign when $\ddot{y}_{02} > \ddot{y}_{01}$.

<u>Example 63</u>. When using a measuring instrument with a zero threshold of $\ddot{y}_{02} = 6.31 \times 10^{-4}$ cm/sec^2, the vibration level of a machine at a given frequency is 105 decibels. What would the vibration level be with respect to a threshold of $\ddot{y}_{01} = 3 \times 10^{-2}$ cm/sec^2?

<u>Solution</u>. The correction is

$$\Delta = 20 \log (6.31 \cdot 10^{-4}) + 31 = -33 \text{ db.}$$

Since $\ddot{y}_{02} < \ddot{y}_{01}$, it is not necessary to change the sign in the correction. The vibration level above the threshold of 3×10^{-2} cm/sec^2 will be

$$\beta_1 = 105 - 33 = 72 \text{ db.}$$

A recently developed special instrument (27) can be conveniently used for vibration and acoustic tests of serially-produced machines and equipment in addition to the ordinary noise gauges and vibration meters. This instrument, called "degree of noise reduction" meter, makes it possible to establish, with a margin of error of 1 decibel, the particular

sound or vibration frequency bands within which the machine does not conform to the prescribed norms. A special scale indicates the percentage of machines tested which do not meet the requirements or the percentage of time within which these requirements are exceeded in particular frequency bands (under conditions of strong pulsating noises).

79. The Major Sources of Error in Measuring Noise and Vibration

Under production conditions, it is quite possible to achieve an accuracy in the measurement of sound and vibration levels of the order of 2 decibels. But this calls for the elimination or diminution of the effects of a number of factors that tend to make the measurements less precise.

One of the frequent sources of error in measuring noise in the open air, in ventilation ducts, in an exhaust gas flow, etc. is the excitation of the sensitive microphone element due to vortex formation around it. By the use of special wind-proof netting and screens (Figure 215) this effect can be greatly reduced, even to zero. The netting is transparent to sound, and at the same time reduces the intensity of the vortices or shifts their sonic spectrum into the ultrasonic frequency region (22). The effectiveness and sound-permeability of the wind-protection device can be checked by comparing the measurements of sound levels when using protected and unprotected microphones in free atmosphere and in the path of an air stream.

When measuring the noise at high sonic frequencies, where the wave-length is comparable to the dimensions of the microphone, the diffraction of the sound waves at the microphone may become a source of error. In view of the reflection of sound from the body of the microphone and the measuring instrument, the sound pressure on the microphone's membrane increases by 4-5 decibels or more. The diffraction distortions can be reduced to a minimum by the use of small microphones placed 10-20 centimeters from the instrument. If this is not possible, the diffraction correction should be established by comparing the readings of the given microphone with those of a "point" microphone in a free sound field.

With small microphones it is possible to avoid another error, namely the error due to directionality of the microphone, i.e., its differing sensitivity in different directions.

Before taking any noise or vibration measurement, one must make sure that there is no interference due to the microphone effect of electronic tubes, which is caused by shaking of the measuring instrument. In the case of a strong vibration of a foundation, which is a frequent

occurrence in the engine rooms of ships, the measuring instrument should be placed on sponge rubber or even held in the hand. To make sure that there is no interference from the microphone effect of tubes, the microphone should be covered with a cap (or the vibration probe removed from the vibrating surface), and a reading of the instrument taken. The microphone effect may occur not only when the instrument itself is shaking, but also in the case of shaking of the cable connecting the vibration probe or microphone to the instrument.

Another significant source of error when working with piezo-electric microphones and vibration pickups is the weak signal at low frequencies due to the high internal resistance of the piezo crystal, and also the weak signal due to damping in the cables at high frequencies. The correction for this (if it is not compensated for by a special feature within the instrument), as well as the corrections for the effect of temperature upon the sensitivity of the piezoelectric elements, are indicated in the descriptions of the instruments and in the instructions for their use.

Errors arising in a noise and vibration analysis with a narrow band analyzer at high sonic frequencies may be due to the uneven rotation of the machine, which produces, as it were, a frequency modulation of the noise. We know from theory (107), that in the case of a small degree of modulation 2F

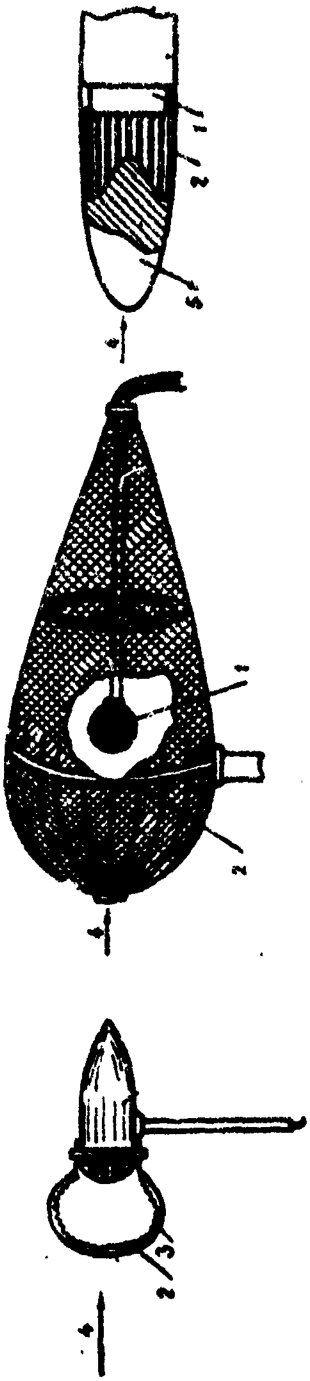


Figure 215. The windproofing of microphones.

1 - microphones; 2 - copper screen; 3 - webbing; 4 - direction of arrival of sound; 5 - fairing.

(the so-called "band of wavering"), the spectrum of a frequency-modulated vibration as well as the spectrum of an amplitude-modulated vibration consists of components of the fundamental frequency f_o and two components ("side bands") with frequencies $f_o + F$ and $f_o - F$.

If the difference in the side bands exceeds the width of the analyzer pass band, both of them fall outside of the sensitivity channel of the given frequency, and the level indicated by the analyzer will be lowered. This level is further reduced with increasing modulation, as a considerable part of the energy is concentrated in components which are still further removed from the fundamental frequency.

Example 64. Determine the possibility of using an analyzer with a pass band width of 4 cps for measuring the level of the tenth harmonic of the noise produced by a ventilator rotating at a speed of 2,400 rpm. The irregularity of the ventilator's rotational speed within a single revolution is 2%.

Solution. The frequency of the tenth harmonic is

$$f = \frac{2,400 \times 10}{60} = 400 \text{ cps.}$$

The band of wavering of the harmonic, $f_k = 0.02f = 8$ cps, exceeds the width of the analyzer pass band, and errors in the analysis are therefore inevitable. For such a condition an analyzer with a constant relative pass band width which is greater than the rotational irregularity, that is greater than 2%, is preferable to the analyzer given in the example.

Just as too large a microphone produces diffraction distortions of the sound field at high frequencies, a vibration pickup with too much mass attenuates vibration by its effect on the vibration surface. The reduction of the vibration amplitude $\Delta\beta$ can be simply expressed by use of the vibration pickup's impedance z_p and the surface's impedance z_s (51):

$$\Delta\beta = 20 \log \left(1 + \frac{z_p}{z_s}\right) \text{ db} \tag{268}$$

The attenuation of vibrations of frequency f, as a function of h, the thickness of an extensive plate supporting a vibration pickup of mass m, is:

$$\Delta\beta = 10 \lg\left(1 + 7{,}5\,\frac{\sqrt{1-\mu^2}}{E\rho}\cdot\frac{m^2 f^2}{h^4}\right) \text{db}, \tag{269}$$

where E, ρ, μ — represent the elastic modulus, density and Poisson's ratio of the plate material.

Obviously, the magnitude of the error is affected principally by the plate thickness. This formula is accurate up to a frequency range of 2,000 cps (5). At higher frequencies, the reduction of the vibration by a vibration pickup of finite mass may be less than the attenuation given by the formula.

With respect to a steel plate, formula (269) gives the following

$$\Delta\beta = 10 \log \left(1 + 4.6 \cdot 10^{-13} \frac{m^2 f^2}{h^4}\right), \tag{270}$$

where m is expressed in grams, h in centimeters.

The formula can be further simplified for the case of higher attenuation values, that is, for greater mass of the pickpup, higher values of frequency and smaller plate thicknesses, as follows:

$$\Delta\beta \cong 20 \log \frac{mf}{h^2} - 123 \text{ db} \tag{271}$$

If a calculation based on this formula produces $\Delta\beta < 6$ decibels, or should even lead to a negative result, it means that formula (270) should be used. In view of the complexity of the interaction process between an actual vibration pickup and the surface, the above formulas, based as they are on an approximate theory, should be used for a quick estimate of the conditions under which a vibration pickup of a given mass can be used, rather than for determining the actual corrections to the vibration pickup's reading.

Example 65. Is it possible to use a vibration pickup weighing 100 grams for measuring the vibration of a 1.5 mm-thick sound isolating housing at frequencies up to 1,000 cps, without any appreciable errors? How far can the error be reduced by measuring the vibration of the same housing with a 10-gram vibration pickup?

Solution. Let us try to determine the pickup's reduction of the housing's vibration at the highest frequency according to formula (271):

$$\Delta\beta \cong 20 \log \frac{100 \cdot 10^3}{(0.15)^2} - 123 = 10 \text{ decibels}$$

We shall do the same with a 10-gram vibration pickup (formula 270):

$$\Delta \beta = 10 \log \left[1 + 4.6 \cdot 10^{-13} \frac{10^2 \times 10^5}{(0.15)^4} \right] = 0.5 \text{ decibels.}$$

Consequently, the "point" vibration pickup will not produce any distortions in the measurements whereas distortions of the vibration spectrum tending to reduce the high-frequency components are inevitable in the case of the 100-gram pickup.

When measuring, by means of pick-ups of finite mass, the average vibration levels in a frequency band, the error cannot be separated from the measured results because of the vibration pickup's reaction upon the vibrating body. The magnitude of this error depends essentially on the nature of the frequency spectrum of the vibrations. A frequency analysis of the vibration is therefore always preferable to measuring the average vibration level.

Ideal as the "point" piezo-vibration pickup may be as a measuring instrument, it may still produce distortions. Distortions can occur at low frequencies when poor-quality filters are used to assist in the measurements, and are occasioned by the extreme sensitivity of a piezo-pickup at high sonic frequencies. Distortions at low frequencies are magnified whenever the noise spectrum contains a large number of high-frequency components.

Let n be an index characterizing the degree of the change in the spectrum of vibrational acceleration with frequency. A spectrum with a 6-decibel rise per octave would then correspond to $n = 1$, a constant frequency spectrum to $n = 0$, a spectrum with a 6-decibel drop per octave to $n = -1$, etc. Let us assume further that Δ_{Φ} is the difference between the sensitivity of some low-frequency filter within its pass band (at frequency f_H) and the "residual" sensitivity of the same filter at the highest operating frequency of the piezo-pickup, f_B. In well-made filters this quantity amounts to 50-60 decibels or more, but in commercial filters it sometimes does not exceed 25-30 decibels. Generally, vibration pickup resonance, enhancing the role of the high-frequency portion of the spectrum, may occur at a frequency of f_B, and also some discrete component of the spectrum may be present. We shall designate the resonance peak amplitude in the characteristics of the vibration pickup by Δ_p, and the amplitude of the discrete component of the vibration spectrum at this frequency by Δ_c. Taking all the foregoing into consideration, it is possible to state the condition for the absence of distortions from the measurements of the vibration level at the frequency f_H in the following form (54):

$$n < \frac{\Delta_\phi - \Delta_p - \Delta_c}{20} \cdot \frac{1}{\log\left(\frac{f_B}{f_H}\right)}. \tag{272}$$

Let us assume that $\Delta_\phi - \Delta_p - \Delta_c = 40$ decibels, which may be close to actual conditions. Then

$$n < \frac{2}{\log\left(\frac{f_B}{f_H}\right)} = \frac{6}{N}, \tag{273}$$

where N represents the number of octaves between the frequencies f_H and f_B.

Example 66. There is a 5 decibel resonance peak (piezo-crystal resonance) in the frequency characteristics of a pickup at its highest operating frequency, $f_B = 13{,}000$ cps. The difference between the sensitivity of the low-frequency filters in their pass band and at the frequency f_B is 45 decibels. Determine the possibility of undistorted measuring of components having a frequency of 25-50 cps for the case of a spectrum characterized by constant amplitudes of the frequency components of vibratory acceleration and for the case of a spectrum with a 5-decibel rise per octave.

Solution. We determine that

$$\Delta\phi - \Delta_p = 45 - 5 = 40 \text{ decibels}$$

Nine octaves are contained in the 25-13,000 cps frequency band. According to formula (273), the following condition should exist:

$$n < \frac{6}{9} = \frac{2}{3}.$$

The case n = 0 corresponds to a spectrum with constant frequency components, and the condition for undistorted measurements in the latter spectrum at low frequencies is therefore fulfilled.

Let us determine the value of n in the spectrum with a 5-decibel rise per octave. We have:

$$20 \log \left(\frac{2f}{f}\right)^n \approx 5,$$

hence $n = \dfrac{1}{4 \log 2} \approx \dfrac{3}{4} > \dfrac{2}{3}$, that is, the above-required condition cannot be achieved.

Considerable distortions could be expected in the latter case because of the effect of the high-frequency portion of the spectrum. A second filter, capable of cutting out the high frequencies, should be used to reduce these distortions. Measurements can also be made at low frequencies (up to 200-300 cps) by a piezo vibration pickup with a relatively low natural frequency (1,000-2,000 cps), above which the sensitivity of such a vibration pickup drops sharply, as a consequence of which the high-frequency portion of the spectrum does not affect the reading. The fact that such vibration pickups have an appreciable mass (see Chapter 4) is not very important in this case, as the reaction of the vibration pickup to a vibrating surface at low frequencies is relatively small (see formula 270). Wide-band, "point"-type, inertia-less piezo-vibration pickups should be used at higher frequencies. At those frequencies it is important to have a reliable contact between the vibration pickup and the surface, and that can be done by screwing it on or gluing it, or, as some authors suggest, by inserting a lead gasket or an oil layer.

80. Calibrating and Checking Noise- and Vibration-Measuring Apparatus

One of the conditions for accurate measurements is the proper tuning and calibrating of the measuring circuits. It is relatively easy to check the electrical circuits of such systems. This is done by means of the calibrating devices located in the various loops of the circuit (amplifiers, recording instruments, analyzers).

Somewhat more complicated is the calibration and checking of the electro-acoustical transducers in the system, namely the microphones and vibration pickups. Theoretically their calibration can be carried out either by removing them from the electrical system and using an electronic voltmeter as an indicator (a step-by-step calibration), or by calibrating them in place in the electrical system (an overall calibration). The latter type of calibration is preferable in practice, as it automatically takes into account the possible accidental mismatch of electrical impedances at the connections between the different elements in the system a matter which affects the sensitivity characteristics of the system as a whole.

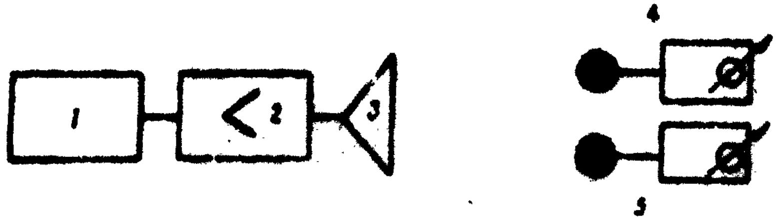


Figure 216. Calibrating a noise-measuring device by the comparison method.

Noise-measuring systems can be calibrated and checked in noise-measuring chambers of the type shown in Figure 21?. Ordinarily used for this purpose is the comparison method (Figure 216), whereby the sound level registered by a device 5, under test, is compared with the level indicated by the calibrating device 4. An output device consisting of a tonal generator 1 (or noise generator and one-third octave filters), amplifier 2 and wide-band loudspeaker 3 is used to produce the sound. The calibrating devices themselves are calibrated at government checking stations by a Rayleigh disc or by primary calibrating microphones.

Stationary vibration test stands as well as portable vibro-plates and mechanical vibration generators are used to calibrate vibration pickups. In contrast to the procedure for noise-measuring systems, one can find the characteristics of absolute sensitivity for vibration-measuring devices without the use of any secondary calibrating instruments. This is done by comparing the values of the parameter to which a pickup placed upon the vibroplate reacts (vibrational velocity or acceleration), with the reading of the pickup. Up to a frequency of several hundred cycles per second, the absolute vibrational amplitudes of the pickup on a vibro-plate can be measured by a direct reading through a microscope. At higher frequencies, use is made of interferometer apparatus or calibration schemes based upon the method of reciprocity (58).

Figure 217. A piston-type vibro-plate for the absolute calibration of vibrometers at low and medium sonic frequencies.

Vibrational devices designed to calibrate vibration meters at sonic frequencies (81, 58) are described in Russian technical periodicals. Figure 217 shows a simple electrodynamic device of that type developed by the author jointly with A. N. Lapenko. The basis of the instrument is a pair of powerful permanent magnets placed one on top of the other. Two sound coils linked by a rigid connection are placed in the clearances between these magnets. Such a vibro-plate system ensures a strictly piston-type of motion, that is, an absence of transverse or rotational displacements. The vibration plate has a massive base, and is installed upon an elastic rubber gasket in such a way that its frequency of free vibration on the gasket lies below the range of operating frequencies.

The vibration pickup which is being tested, visible in the Figure, is screwed on to the plate above the upper sound coil. The vibrations of the pickup can be observed through a microscope focused on one of the shiny dots on the lateral surface of the probe (such dots can be made by touching the surface with fine emery cloth). When the vibration plate is excited, the luminous dots, as seen through the microscope's eyepiece, stretch out into bands or narrow ellipses whose vertical dimension, indicated on the eyepiece scale, corresponds to the double amplitude of the pickup's vibrations. A reading of the pickup's meter can be taken at the same time, or -- if the sensitivity of a separately installed vibration pickup is being measured -- a reading of a high-resistance electronic voltmeter connected to the vibration pickup can be taken. The above-described simple instrument for calibrating vibration pickups up to a frequency of 400-500 cycles per second can easily be produced in any plant laboratory.

The sensitivity of noise- and vibration-measuring devices must be controlled not only during the preparations for measurements but also during the course of such measurements.

When using noise-measuring devices, such control can be maintained by the introduction of standard sources of sound (pistonphones, "ball clocks") for checking the noise gages.

A similar principle of comparison with a more or less constant source of vibration can be applied also to vibration-measuring devices. A well-isolated metal plate on which is installed a hammer vibrator with a constant rpm of its driver could be used as a standard source of sonic vibration. A standardized vibration pickup, securely fastened to a precisely fixed spot on the metal plate and not participating in the measurements, can be used to maintain additional control over the vibration level. The readings of that vibration pickup can be compared with the readings of the working pickups when they are periodically placed on the plate (also at precisely fixed points).

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pages 1-300: 5200
pages301-414: 5132
CSO: 8092-D

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